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MODELING AND CONTROL OF A NOVEL
ROBOTIC ACTUATOR

by

John David Ingram Jr.

December 1988

Thesis Advisor: David L. Smith

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This thesis documents analysis of the feasibility of such an actuator. Computer modeling and simulation are accomplished. A hardware test bed with a microcomputer control and parameter sensing interface is designed and constructed for the purpose of model validation and demonstrations. Automatic control software is designed and implemented on the test bed, and performance evaluations are made. From the observations made during the analysis process, design recommendations are formulated and proposed.

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Modeling and Control of a Novel Robotic Actuator

by

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Submitted in partial fulfillment of the
requirements for the degree of

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ABSTRACT

With the increased use of robots in industry and the military, new robot-specific actuators will be developed to better meet functional requirements. One concept to be considered is a stiff pneumatic-hydraulic actuator for mobile anthropomorphic robot application.

This thesis documents analysis of the feasibility of such an actuator. Computer modeling and simulation are accomplished. A hardware test bed with microcomputer control and parameter sensing interface is designed and constructed for the purpose of model validation and demonstrations. Automatic control software is designed and implemented on the test bed, and performance evaluations are made. From the observations made during the analysis process, design recommendations are formulated and proposed.

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TABLE OF NOMENCLATURE

A	area
α	linkage geometry angle
β	bulk modulus; linkage geometry angle
C_d	orifice discharge coefficient
d	diameter
ΔP	pressure difference
F	force
J	mass moment of inertia
k	mechanism stiffness
ℓ	length
m	mass
ϕ	linkage geometry angle
P	pressure
η	linkage geometry angle
Q	volumetric flow rate
ρ	density
t	time
T	temperature
θ	rotational coordinate
V	volume
W	weight
X	linear coordinate

Subscripts

a	atmosphere
F	foundation
H	hydraulic
L	linkage
oh	hydraulic orifice
P	pneumatic
ph	hydraulic piston
pp	pneumatic piston
r	rod
rh	hydraulic rod
rp	pneumatic rod
x	linkage coordinate
y	linkage coordinate
0	initial
1	coordinate 1; air supply
2	cylinder chamber 2; coordinate 2
3	cylinder chamber 3
4	cylinder chamber 4
5	cylinder chamber 5
11	linkage dimension
12	linkage dimension
13	linkage dimension
14	linkage dimension

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I. INTRODUCTION

Robotic applications have increased steadily with advancing technology. Installation of manipulators to automate assembly line tasks has become commonplace because of the increased productivity, reliability, and cost savings which can be realized through their use. Projections estimate the value of the industrial robot population worldwide to be 10 billion dollars by the year 1990 [Ref. 1]. The military is also interested in the application of robotics to missions where they may decrease risks to personnel and significantly enhance the probability of mission completion [Ref. 2].

A divergence in requirements between commercial and military robotic devices occurs because of the military need for mobility. While industrial manipulators are most often used in stationary installations such as factories, the nature of military missions dictate that a manipulator must be able to go to the location where its task will be accomplished, instead of having the job brought into its operating envelope. The mobility issue brings with it additional requirements for manipulator subsystems which are usually not considered in a stationary design. Some of these are power supply weight, system endurance, and ruggedness in changing environments.

Manipulator actuators must generally satisfy a larger number of functional criteria than similar power delivery devices used in other systems. In addition to efficiency and reliability, the desirable qualities of the actuators for a robot arm may include high torque or force output throughout translation, quick response to signal orders, smooth reversibility, high stiffness with low power consumption when idle, positioning accuracy, and any other characteristics which are dictated by the functions which drive the total system design. Traditional choices for actuators have been confined mainly to electric motors and either hydraulic or pneumatic motors and cylinders. While each actuator category has been used successfully in manipulator applications, satisfactory conformity to the list of qualities needed is often difficult to achieve.

An innovative actuator type which may conform better to manipulator system requirements, particularly in mobile applications, is the stiff pneumatic-hydraulic actuator [Ref. 3]. Figure 1 is a notional diagram of the construction of such an actuator in cross section, suitable for use in a low power, light load application [Ref. 4]. The main motive power for this actuator would be provided by a low pressure compressed air source. In a mobile arm, this could conceivably be a rechargeable high pressure air tank with a regulator. The closed loop hydraulic side is a computer controlled, self-contained circuit which adds

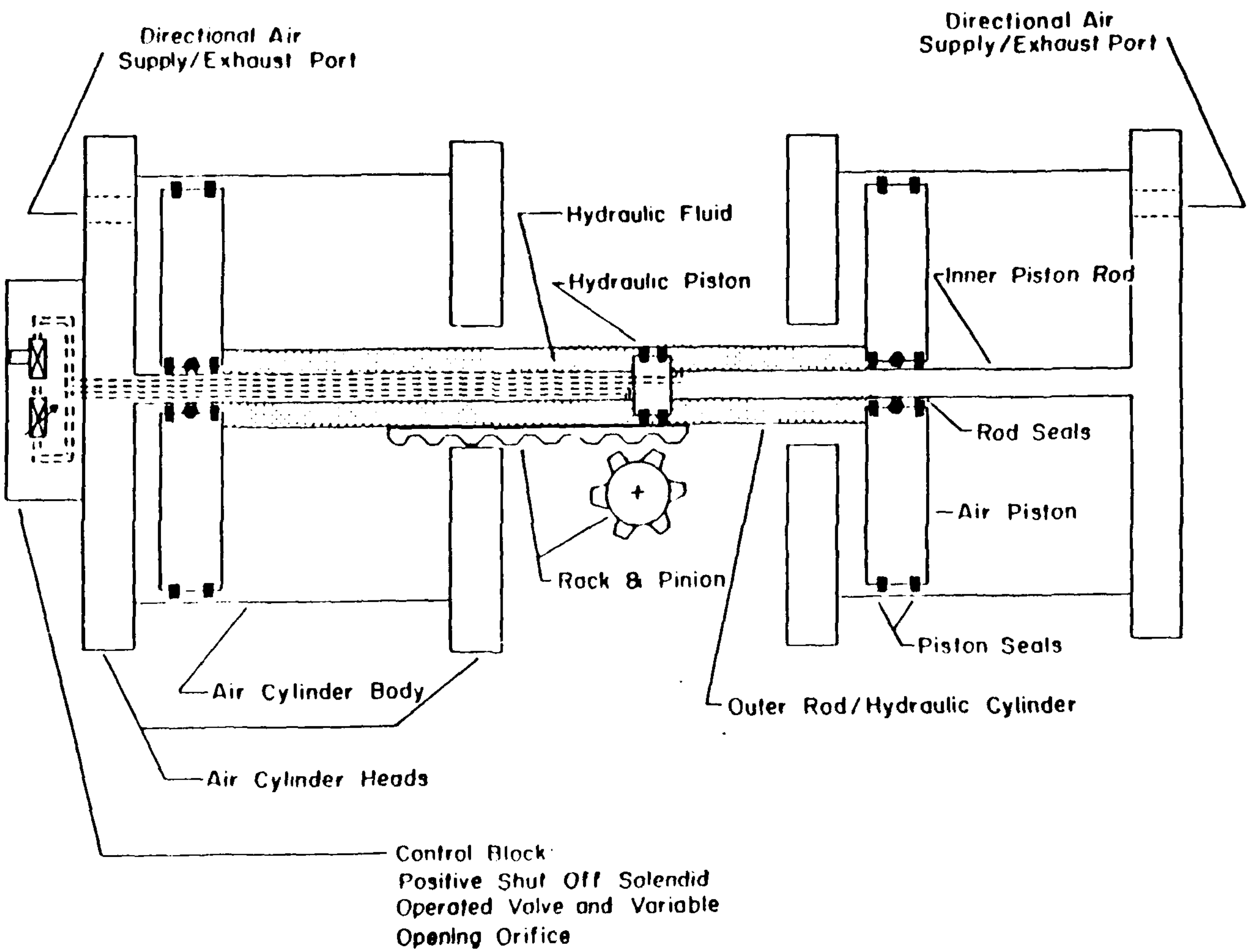


Figure 1. Conceptual Novel Actuator

motion dampening for precise position control during translation--a feature that pneumatic cylinders do not normally have. When ordered position is reached, the hydraulics add high stiffness, again an unusual feature for a low pressure pneumatic actuator.

To investigate the suitability of this actuator for use on a mobile operational manipulator, a concept feasibility was conducted using the systems analysis approach and the functional-based design principles detailed by Blanchard and Fabrycky [Ref. 5]. Following the concept analysis, a first order mechanical system analysis was developed which was used to perform system simulations to gather preliminary motion prediction data. The results of this conceptual design work are recorded in Reference 6.

Components of an operating hardware prototype which was used for concept demonstration were received from the Naval Ocean Systems Center. Figures 2 and 3 show the components mounted on a rigid benchtop base. The Novel Actuator Test Bed was created from this apparatus through structural modifications and addition of computer controllable hardware, sensors, and interface circuits.

The fundamental purpose of this thesis was to demonstrate the viability of the novel actuator concept by developing and demonstrating a controller which would deliver satisfactory performance. Performance characteristics were developed [Ref. 6] and used as the

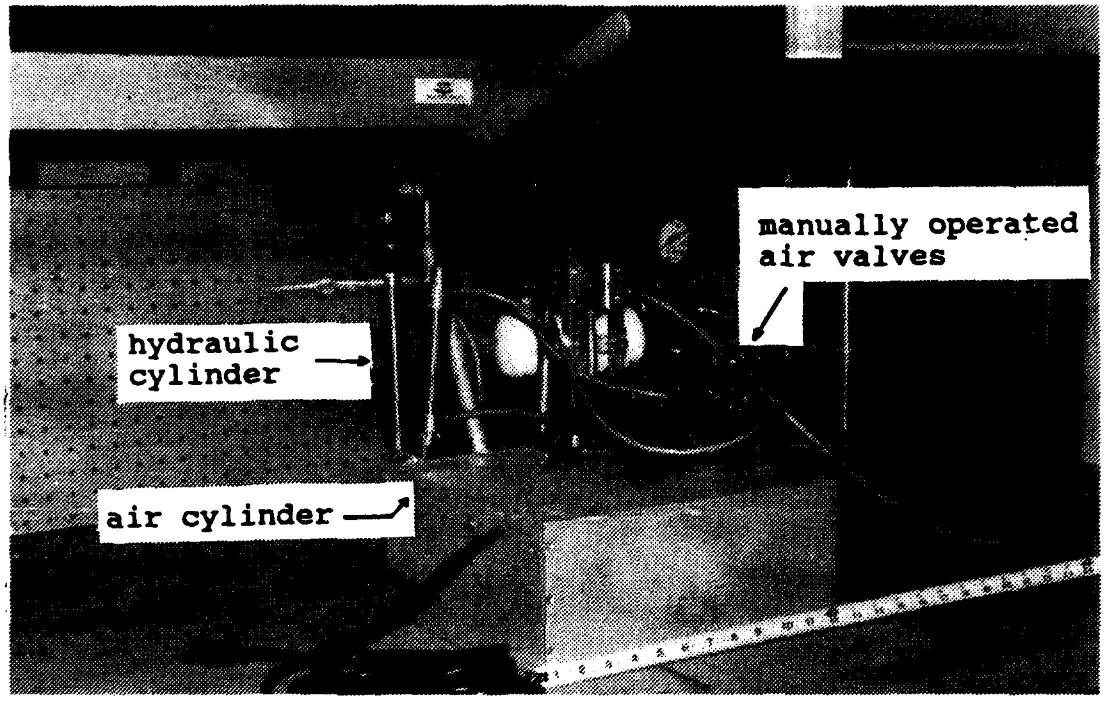


Figure 2. Demonstration Model

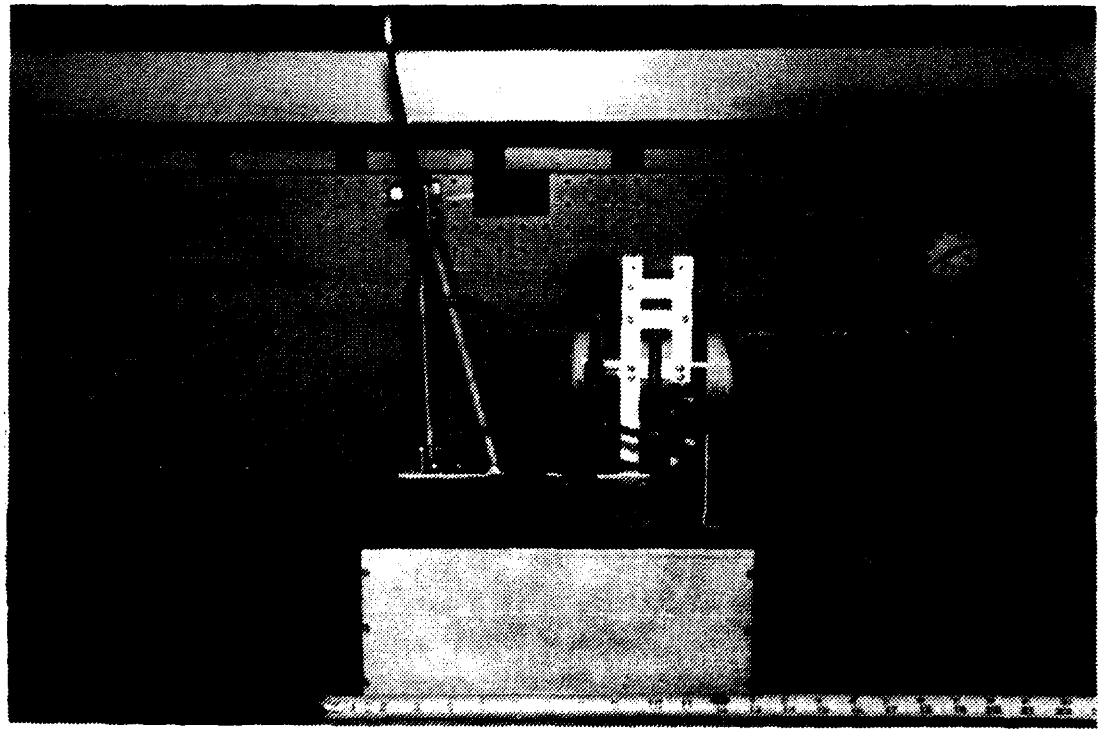


Figure 3. Demonstration Model

design criteria for the control scheme which was implemented on the Novel Actuator Test Bed. The primary characteristics were:

1. The ability to move and position a maximum five pound payload.

2. The ability to achieve accuracy in placement of ±0.20 inches from ordered position.

3. The ability to meet a point-to-point placement time within the operating envelope of no more than three seconds.

The general design procedures used in the thesis were:

1. Develop a mathematical model description of the system from applicable theory.

2. Verify that the model provided a suitable description of system dynamics.

3. Use model prediction data to design a position control system meeting the above guidelines for performance.

4. Implement the controller and check compliance with the desired characteristics.

II. MODELING AND SIMULATION

Description of the Novel Actuator Test Bed system for any achievable status was the goal of design modeling. This was completed by performing two separate analyses. An estimate of mechanism stiffness was acquired through analysis of rest conditions. Analysis of dynamic conditions enabled the programming of a numerical simulation which could predict system motion.

A. STIFFNESS ANALYSIS

A simplified model of the Novel Actuator Test Bed was constructed as shown in Figure 4. The first objective was to gain a measure of the system stiffness when the joint was stationary. In conducting the stiffness analysis, it was assumed that the mechanism was in the position shown in Figure 4, that the contribution of the air cylinder to stiffness was a secondary effect, and that the hydraulic cylinder was in mid stroke, as shown in Figure 5. With the air cylinder deenergized and the hydraulic solenoid valve closed, the system was motionless. The stiffness of the mechanism was characterized in terms of pounds per inch of deflection away from the no-load position when a load was applied to the linkage. Consequently, assuming that the foundation and linkage materials were sufficiently rigid so that structural deflection was negligible, the system

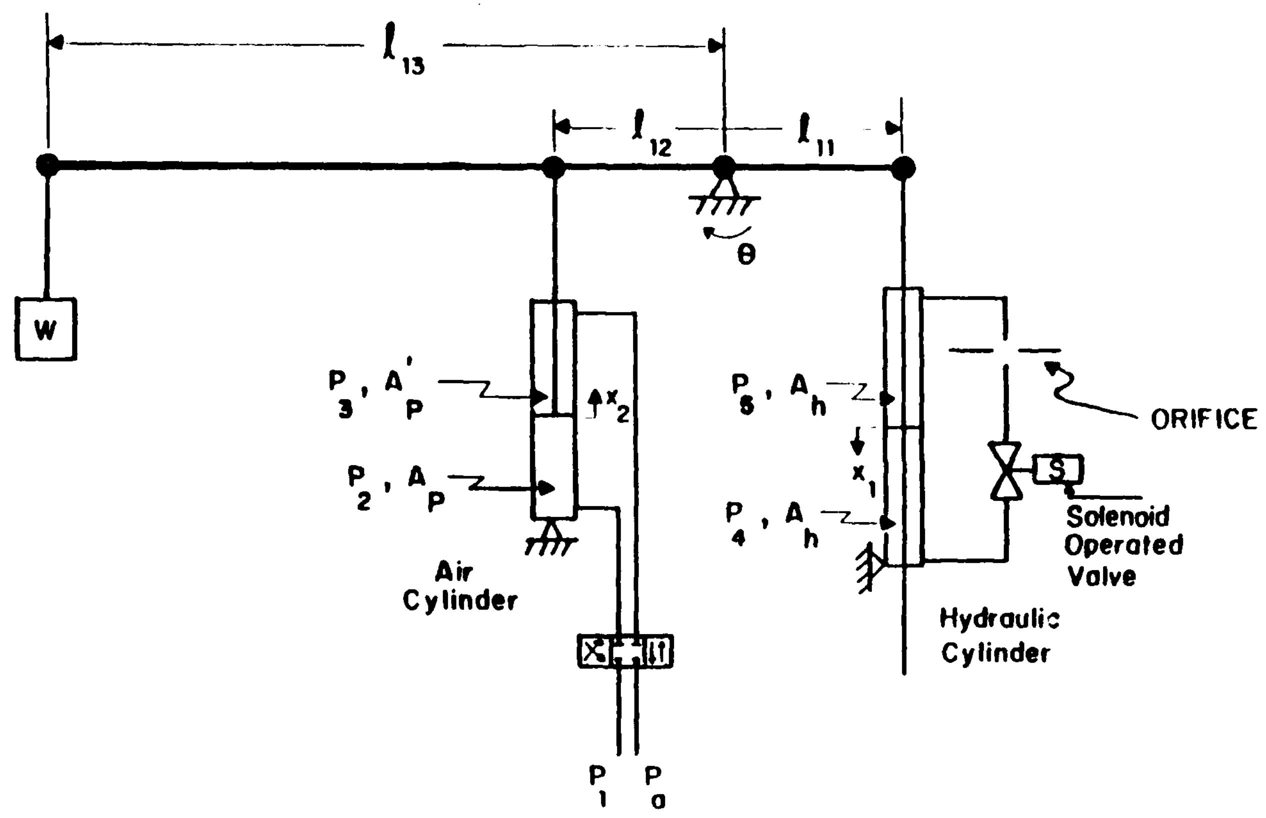


Figure 4. Simplified System Model

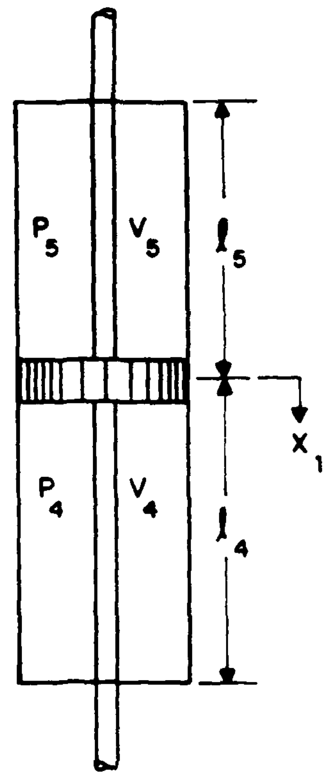


Figure 5. Hydraulic Cylinder Model

stiffness was a function of the force of the load transmitted to the hydraulic cylinder, and the bulk modulus of the hydraulic fluid. A summation of moments about the pivot point (Figure 4) gave:

$$\Delta P_H A_H \ell_{11} = W\ell_{13} \tag{1}$$

In turn, the pressure difference across the hydraulic cylinder was expressed as:

$$\Delta P_H = \frac{W\ell_{13}}{A_H \ell_{11}} \tag{2}$$

Since the rotary deflection of the mechanism could be expressed in terms of the linear deflection of the hydraulic piston, X_1, further stiffness analysis focused on the hydraulic cylinder as modeled in Figure 5. The assumption was made that flow past the rod seals and piston seals was negligible. These leakage flows would be the only flows possible for this model, so:

$$Q = 0 \tag{3}$$

The mass of fluid in each chamber of the cylinder was:

$$m = \rho V \tag{4}$$

Applying the continuity equation to the chambers:

$$\frac{dm}{dt} = \frac{d(\rho V)}{dt}$$

or

$$\frac{dm}{dt} = \rho \frac{dV}{dt} + V \frac{d\rho}{dt} \qquad (5)$$

The bulk modulus was defined as:

$$\beta \equiv \rho_o \left(\frac{\partial P}{\partial \rho}\right)_T \qquad (6)$$

Assuming that the pressure of the fluid was independent of temperature gave:

$$\beta \equiv \rho_o \frac{dP}{d\rho} \qquad (7)$$

Substituting this into Equation (5):

$$\frac{dm}{dt} = \rho \frac{dV}{dt} + V \frac{d\rho}{dt}\left(\rho \frac{dP}{d} \frac{1}{\beta}\right)$$

or

$$\frac{dm}{dt} = \rho \frac{dV}{dt} + \frac{\rho V}{\beta} \frac{dP}{dt} \qquad (8)$$

Dividing this resultant by ρ gave an equation for Q:

$$Q = \frac{dV}{dt} + \frac{V}{\beta} \frac{dP}{dt} \qquad (9)$$

The assumption that Q = 0 yielded:

$$\frac{dV}{dt} = - \frac{V}{\beta} \frac{dP}{dt} \qquad (10)$$

For each chamber of the hydraulic cylinder:

$$\frac{dV_4}{dt} = - \frac{V_4}{\beta} \frac{dP_4}{dt} , \quad \frac{dV_5}{dt} = - \frac{V_5}{\beta} \frac{dP_5}{dt}$$

or

$$\frac{dV_4}{V_4} = - \frac{1}{\beta} dP_4 , \quad \frac{dV_5}{V_5} = - \frac{1}{\beta} dP_5 \qquad (11)$$

Integrating both sides of each equation:

$$\ln(\frac{V_4}{V_{04}}) = - \frac{1}{\beta}(P_4-P_{04}) , \ln(\frac{V_5}{V_{05}}) = - \frac{1}{\beta}(P_5-P_{05}) \qquad (12)$$

Then, assuming:

$$P_{04} = P_{05} = P_0 \; , \qquad V_{04} = V_{05} = V_0$$

(thus $\ell_4 = \ell_5 = \ell$ as in Figure 5). Solving each equation for P_0 then combining yielded an expression for the pressure difference across the piston:

$$P_0 = P_4 + \ln\left(\frac{V_4}{V_0}\right)\beta \; , \quad P_0 = P_5 + \ln\left(\frac{V_5}{V_0}\right)\beta$$

$$P_4 + \ln\left(\frac{V_4}{V_0}\right)\beta = P_5 + \ln\left(\frac{V_5}{V_0}\right)\beta$$

$$P_4 - P_5 = \beta \ln\left(\frac{V_5}{V_4}\right) \tag{13}$$

$$V_5 = A(\ell + X_1) \; , \quad V_4 = A(\ell - X_1) \tag{14}$$

$$P_4 - P_5 = \Delta P_H = \beta \ln\left(\frac{A(\ell + X_1)}{A(\ell - X_1)}\right)$$

$$\Delta P_H = \beta \ln\left(\frac{1 + X_1/\ell}{1 - X_1/\ell}\right) \tag{15}$$

Solving for linear deflection X_1 and substituting for ΔP_H using Equation (2) gave:

$$X_1 = \left(\frac{\exp(\Delta P_H/\beta) - 1}{\exp(\Delta P_H/\beta) + 1}\right) \ell$$

$$X_1 = \left(\frac{\exp(W\ell_{13}/A_H\ell_{11}\beta) - 1}{\exp(W\ell_{13}/A_H\ell_{11}\beta) + 1}\right) \ell \qquad (16)$$

Stiffness was assigned a value, k, which followed the equation:

$$W = k\ X_1$$

or

$$k = \frac{W}{X_1} \qquad (17)$$

Substituting the value for X_1 from Equation (16) yielded:

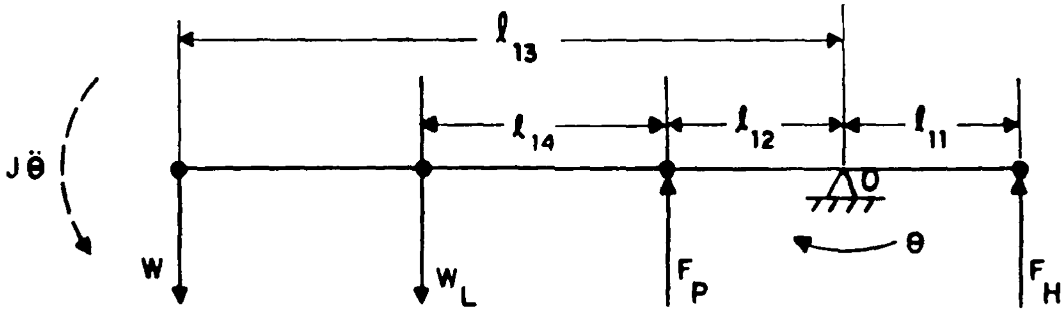
$$k = \frac{W}{\left(\frac{\exp(W\ell_{13}/A_H\ell_{11}\beta) - 1}{\exp(W\ell_{13}/A_H\ell_{11}\beta) + 1}\right) \ell} \qquad (18)$$

An effective bulk modulus for the system can be found using the method described by Merritt [Ref. 7]. A small volume of trapped air within the hydraulic system can be seen to greatly reduce the bulk modulus and thus the stiffness of the mechanism. A representative calculation of volume of trapped air within the hydraulic system can be seen to greatly reduce the bulk modulus and thus the stiffness of the mechanism. A representative calculation of stiffness for the mechanism was made. It was assumed that there was no entrained air in the system. A value for the bulk modulus of 220,000 psi [Ref. 7] for petroleum based hydraulic fluids was used in Equation (18), along with appropriate measured dimensions and a load value of five pounds. The stiffness of the mechanism was calculated to be 10,242 pounds per inch.

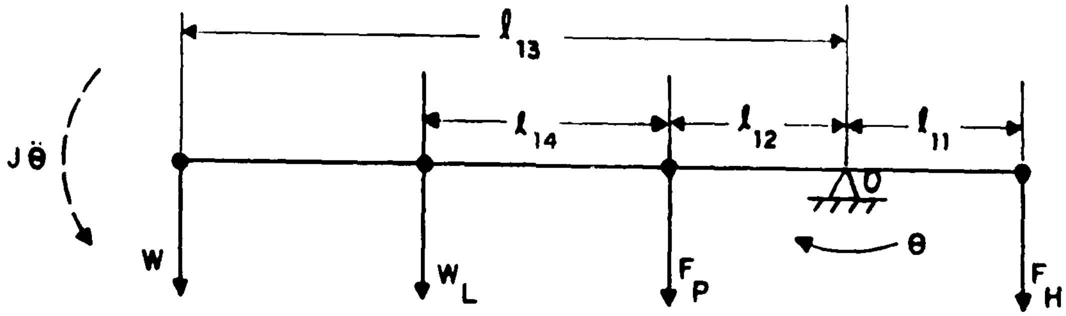
B. DYNAMIC MOTION ANALYSIS

Some refinement was necessary to improve the system dynamic model developed [Ref. 6] using first order approximations. Equations of motion for both raising and lowering motions of the mechanism were desired which would more closely represent actual system behavior, and deliver acceptable accuracy of the system model. The most significant model improvement was to account for changes in the linkage geometry throughout the entire range of motion. The forces on the linkage were then resolved into components, and those components not contributing to system dynamics were discarded.

Forces acting on the linkage for both motions are shown in Figure 6. A similar analysis was used for each case to obtain the equation of motion [Ref. 8]. The following presentation is for the case of a raising motion.



A. Raising Motion



B. Lowering Motion

Figure 6. Forces Acting on the Linkage

A summation of moments about point o in Figure 6A resulted in the following equation:

$$-J\ddot{\theta} - W\ell_{13} - W_L(\ell_{12} + \ell_{14}) - F_H\ell_{11} + F_P\ell_{12} = 0$$

or

$$F_p\ell_{12} - F_H\ell_{11} - W\ell_{13} - W_L(\ell_{12} + \ell_{14}) = J\ddot{\theta} \qquad (19)$$

As a point of reference, the zero position for the rotational coordinate θ was defined to be when the linkage bar was parallel to the base structure, as depicted in Figure 4. Further, a clockwise motion from this position would describe a positive value of θ. Consequently, when the linkage was away from the zero position, only a portion of the forces contributing to the moments in Equation (19) were effective in causing system motion. Relative coordinates x and y, with x being parallel to the long axis of the linkage bar and y being perpendicular to the long axis, were chosen to help separate the forces into components. As illustrated in Figure 7, only the y components of the load and cylinder forces were important to system dynamics. The revised system equation was:

$$F_{py}\ell_{12} - F_{Hy}\ell_{11} - W_y\ell_{13} - W_{Ly}(\ell_{12} + \ell_{14}) = J\ddot{\theta} \qquad (20)$$

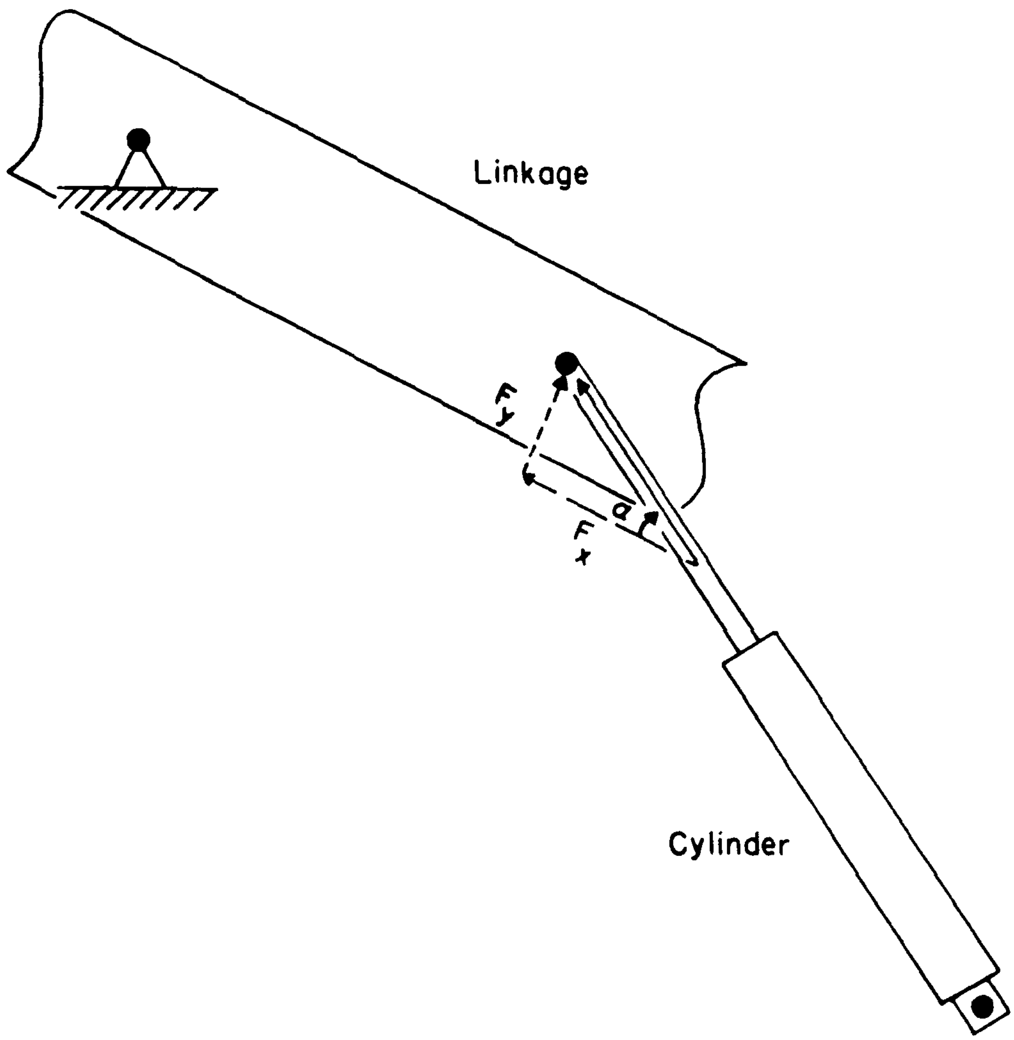


Figure 7. Typical Cylinder Force Components
(Angular Relations Exaggerated)

The active components of the load terms depended only upon coordinate θ. So:

$$W_y = W \cos \theta \ , \qquad W_{Ly} = W_L \cos \theta \tag{21}$$

To solve for the y components of the cylinder forces, it was necessary to utilize a variable angle α, shown in Figure 7, which was a function of the rotational coordinate θ. Figure 8 illustrates the additional mechanism geometry necessary to solve for α. The active cylinder components were:

$$F_{Py} = F_P \sin \alpha_P \tag{22}$$

$$F_{Hy} = F_H \sin \alpha_H \tag{23}$$

The foundation angles β_P and β_H were calculated from fixed lengths:

$$\beta_P = \cos^{-1} \left(\frac{\ell_{12}}{\ell_{FP}}\right) \tag{24}$$

$$\beta_H = \cos^{-1} \left(\frac{\ell_{11}}{\ell_{FH}}\right) \tag{25}$$

The variable angles ϕ_P and ϕ_H were defined as:

$$\phi_P = \beta_P + \theta \tag{26}$$

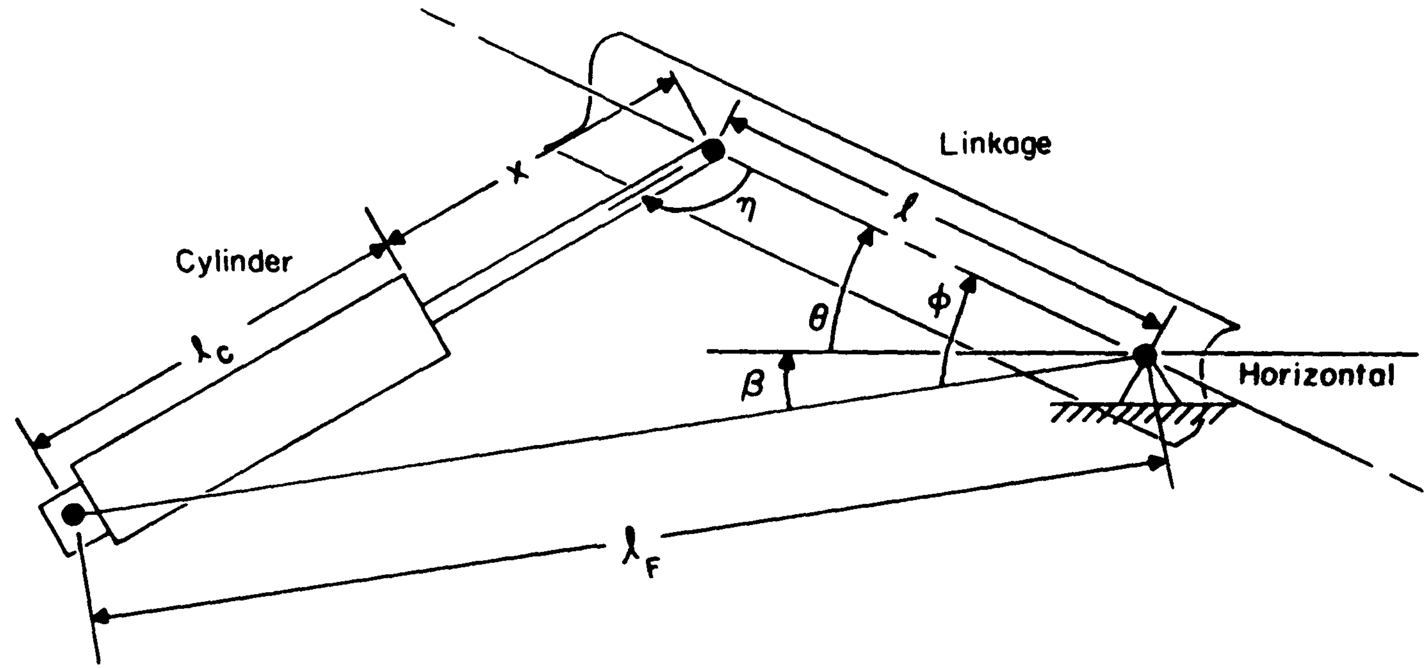


Figure 8. Typical Linkage Geometry (Angular Relations Exaggerated)

$$\phi_H = \beta_H - \theta \tag{27}$$

Trigonometric relations were then used to solve for the supplementary angles, η_P and η_H, to the respective α angles, and enabled solutions for α_P and α_H:

$$\ell_P + X_2 = (\ell_{FP}^2 + \ell_{12}^2 - 2\ell_{FP}\ell_{12} \cos \phi_P)^{1/2} \tag{28}$$

$$\ell_H - X_1 = (\ell_{FH}^2 + \ell_{11}^2 - 2\ell_{FH}\ell_{11} \cos \phi_H)^{1/2} \tag{29}$$

$$\eta_P = \sin^{-1}[(\frac{\ell_{FP}}{\ell_P + X_2}) \sin \phi_P] \tag{30}$$

$$\eta_H = \sin^{-1}[(\frac{\ell_{FH}}{\ell_H - X_1}) \sin \phi_H] \tag{31}$$

$$\alpha_P = \pi - \eta_P \tag{32}$$

$$\alpha_H = \pi - \eta_H \tag{33}$$

The remaining task in the dynamic analysis was to cast all the terms of Equation (20) in a common variable coordinate. The rotational coordinate θ was chosen, and the cylinder forces were examined to perform any necessary coordinate transformations.

Figure 9 shows the forces acting on the air piston. The piston areas were:

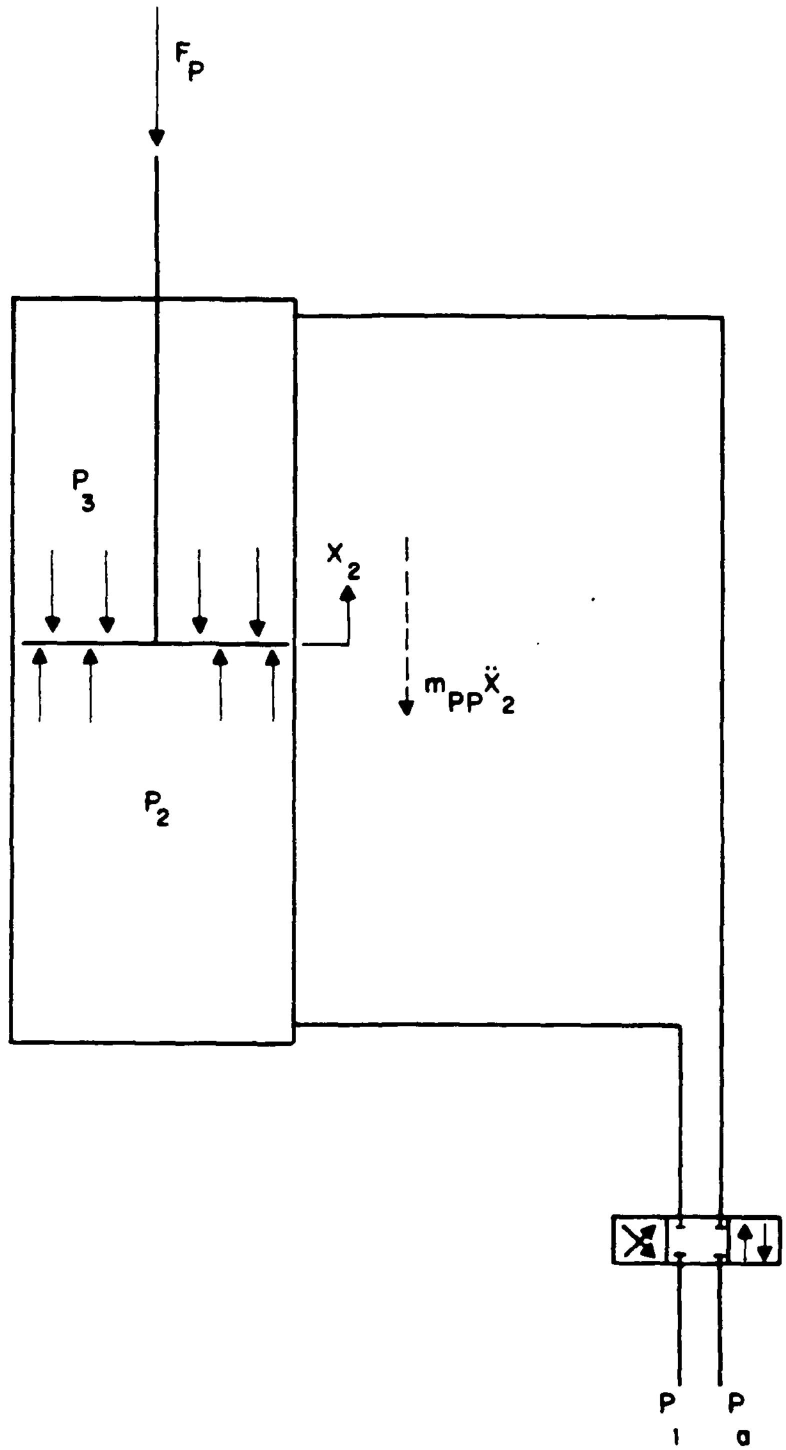


Figure 9. Forces Acting on the Air Piston

$$A_p = \frac{\pi d^2}{4} \tag{34}$$

$$A_p' = A_{pp} - A_{rp} \tag{35}$$

where:

$$A_{rp} = \text{rod area} = \frac{\pi dr^2}{4}$$

So summing the forces yielded:

$$A_p P_2 - A_p' P_3 - F_p - m_{pp} \ddot{X}_2 = 0 \tag{36}$$

Assuming the mass of the piston was negligible:

$$F_p = A_p P_2 - A_p' P_3 \tag{37}$$

Volume flow equations for the air cylinder and control valve [Ref. 9] were used to size the control valve to exceed the flow required to meet the design speed criteria, as discussed in Reference 6. This allowed the assumption that:

$$F_p \cong A_p P_1 \tag{38}$$

Thus, for a given supply pressure, the force delivered by the air piston was specified.

Figure 10 shows the forces acting on the hydraulic piston. The area of the hydraulic piston was:

$$A_H = \frac{\pi d^2}{4} - A_{rh} \tag{39}$$

Summing the forces as with the air piston gave:

$$F_H + A_H P_5 - A_H P_4 - m_{ph} \ddot{X}_1 = 0 \tag{40}$$

Assuming negligible piston mass yielded:

$$F_H = A_H (P_4 - P_5) \tag{41}$$

Contrary to the air cylinder case, the pressures in the oil cylinder were not known. A continuity analysis for the oil cylinder provided a way to determine the pressure drop across the piston for a given operating speed and orifice area. The orifice in the hydraulic line connecting the cylinder ports was treated as a circular orifice through which flow was turbulent. The following equation for volumetric flow through the orifice taken from Merritt [Ref. 7] was applicable:

$$Q_H = C_d \, A_{oh} \, ((2/\rho)(P_4 - P_5))^{1/2} \tag{42}$$

where:

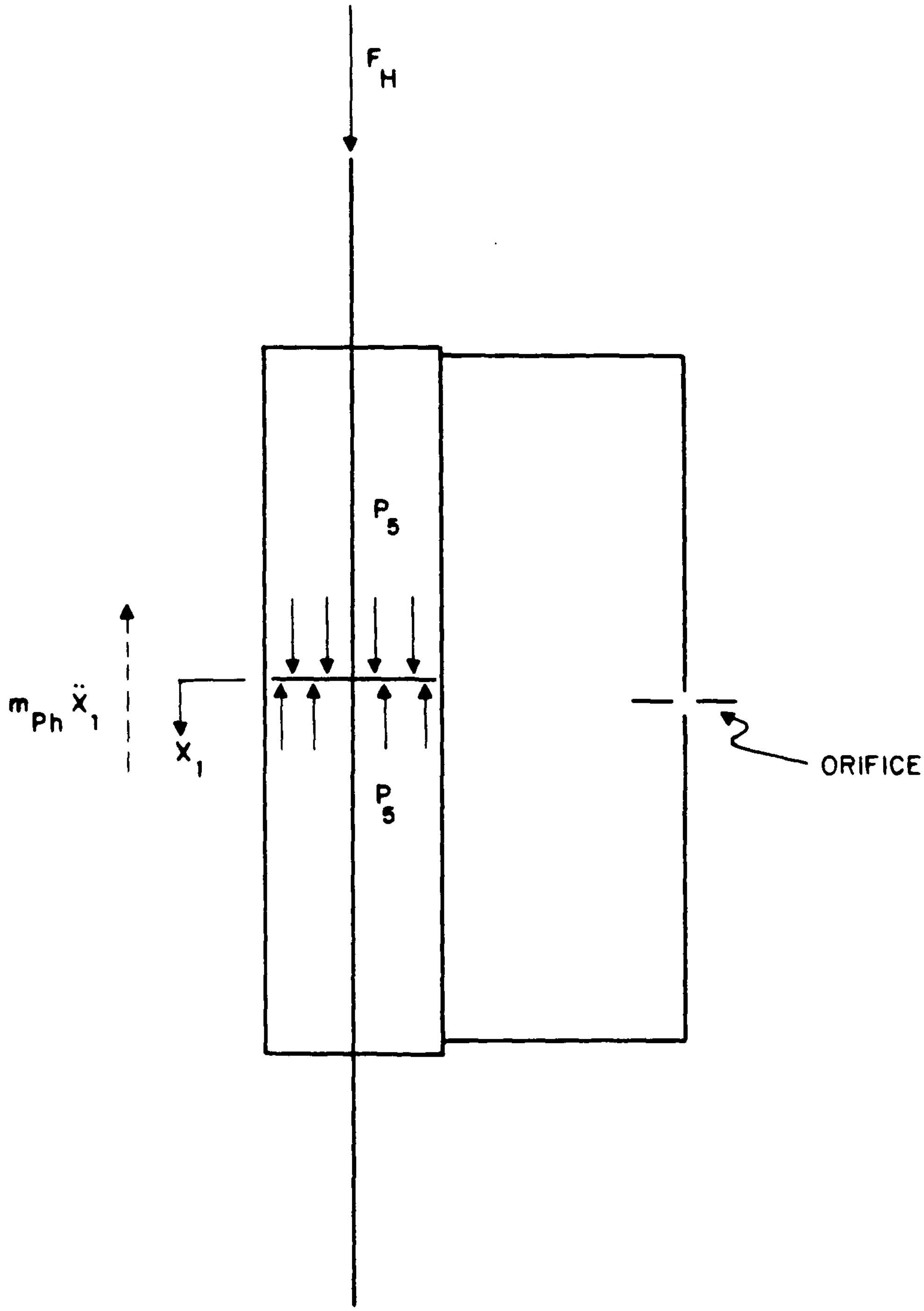


Figure 10. Forces Acting on the Hydraulic Piston

Q_H = volumetric flow, in^3/sec,

C_d = orifice discharge coefficient,

A_{oh} = orifice area in square inches,

ρ = density, $lb\text{-}in^2/sec^4$,

P_4 = chamber 4 pressure, psig,

P_5 = chamber 5 pressure, psig.

By continuity, flow into chamber 5 equaled flow out of chamber 4, so:

$$Q_4 = Q_5 = A_H \dot{X}_1 \tag{43}$$

Also:

$$Q_4 = A_5 = A_H \tag{44}$$

Combining Equations (42) and (43) and solving for the hydraulic pressure difference gave:

$$P_4 - P_5 = \frac{\rho}{2}\left(\frac{A_H \dot{X}_1}{C_d A_{oh}}\right)^2 \tag{45}$$

Transformation of this expression into terms of θ was achieved by solving the combination of Equations (27) and (29) for X_1, taking the derivative of the result, and substituting into Equation (45):

$$X_1 = \ell_H - [\ell_{FH}{}^2 + \ell_{11}{}^2 - 2\ell_{FH}\ell_{11}\cos(\beta_H - \theta)]^{1/2} \tag{46}$$

$$\dot{X}_1 = \ell_{FH}\ell_{11}\dot{\theta}\sin(\beta_H-\theta)[\ell_{FH}^2+\ell_{11}^2-2\ell_{FH}\ell_{11}\cos(\beta_H-\theta)]^{-1/2} \quad (47)$$

$$P_4-P_5 = \frac{\rho}{2}\left(\frac{A_H\ell_{FH}\ell_{11}\dot{\theta}\sin(\beta_H-\theta)}{C_dA_{oh}[\ell_{FH}^2+\ell_{11}^2-2\ell_{FH}\ell_{11}\cos(\beta_H-\theta)]^{1/2}}\right)^2 \quad (48)$$

The hydraulic cylinder force was then put into a form dependent upon rotational coordinate θ:

$$F_H = \frac{A_H\rho}{2}\left(\frac{A_H\ell_{FH}\ell_{11}\dot{\theta}\sin(\beta_H-\theta)}{C_dA_{oh}[\ell_{FH}^2+\ell_{11}^2-2\ell_{FH}\ell_{11}\cos(\beta_H-\theta)]^{1/2}}\right)^2 \quad (49)$$

Equation (20) represented the system equation of motion. The supporting equations necessary to define the terms of this expression in a common coordinate were Equations (21) through (33) and Equations (38) and (49). A numerical simulation program incorporating these equations was written for use with the Dynamic Simulation Language (DSL) software environment available on the IBM 3033 mainframe computer network [Ref. 10]. DSL used a Runge-Kutta equation solving routine to solve the expression for θ and its derivatives. The model simulation was then exercised to obtain motion predictions of mechanism position and velocity versus time in response to given inputs of air supply pressure (P_1), applied load (W), and hydraulic modulating orifice opening (A_{oh}). A listing of the DSL programs used to predict response for both raising and lowering motions can be found in Appendix A.

III. MODEL VALIDATION

A hardware platform was necessary in order to verify that the theoretical model provided an acceptable forecast of an actual system's motion. Therefore, the original concept demonstration model was modified to enable computer control and data acquisition. Experimental data runs were then made for a full range of load conditions and hydraulic control orifice openings. Finally, comparison of actual and theoretical data was made to determine model validity.

A. TEST BED CONSTRUCTION

Construction of the Novel Actuator Test Bed allowed computerized operation and monitoring of the concept demonstration hardware model for the purposes of actual data collection to validate the theoretical model predictions. It also provided the apparatus needed in order to implement a position controller and determine its performance. The following list contains descriptions of the major components of the Novel Actuator Test Bed:

1. Air Cylinder: Clippard Minimatic UDR-173. 1 1/16 inch bore, 3 inch stroke, 5/16 inch rod diameter.

2. Hydraulic Cylinder: Clippard Minimatic H9D-3D. 9/16 inch bore, 3 inch stroke, 1/4 inch rod diameter.

3. Air Cylinder Control Valve: Koganei, Ltd. 110 series, 5 port 3 position closed center, solenoid operated, 24V DC. Rated flow of 14 scfm at 100 psig supply pressure with a response time of 30 milliseconds.

4. Hydraulic Orifice Control Valves: Skinner Precision Industries B series, miniature 3 port, solenoid operated, 24V DC. Rated 150 psi with 3/64 inch passage diameter and a response time of 4 to 8 milliseconds.

5. Position Sensing Device: Sumtak model LBL incremental optical shaft encoder, 2048 pulses per revolution.

6. Computer: IBM PC XT with 8087 math coprocessor.

7. Computer-Sensor Interface: Fischer Computer Systems SEC-PC shaft encoder counter interface board, 4 state change counts per input pulse, binary counter chips, fully addressable TTL input and output ports capable of being used for component control.

8. Computer-Valve Interface: In-house manufactured solenoid valve interface board and 24V DC power supply, one Crydom 6311 opto-isolated SSR output module per solenoid, capable of actuating valve solenoid in response to a TTL logic signal through the output port of the SEC-PC board. A detailed schematic of the solenoid valve interface is included as Appendix B.

The components of the test bed were mounted on and within the foundation base structure as shown in Figures 11 and 12. Data transmissions to and from the computer were made through a 50 pin "D" connector to the SEC-PC board installed in an expansion slot. Control of fluid flow between the two chambers of the hydraulic cylinder was accomplished by a parallel arrangement of four different fixed size, solenoid valve operated orifices. The size for the smallest orifice was chosen based upon speed predictions of the first order approximation system analysis done in Reference 6. Each subsequent orifice was sized using the criteria of a two-to-one increase in area, so that the fourth orifice had an area eight times that of the smallest

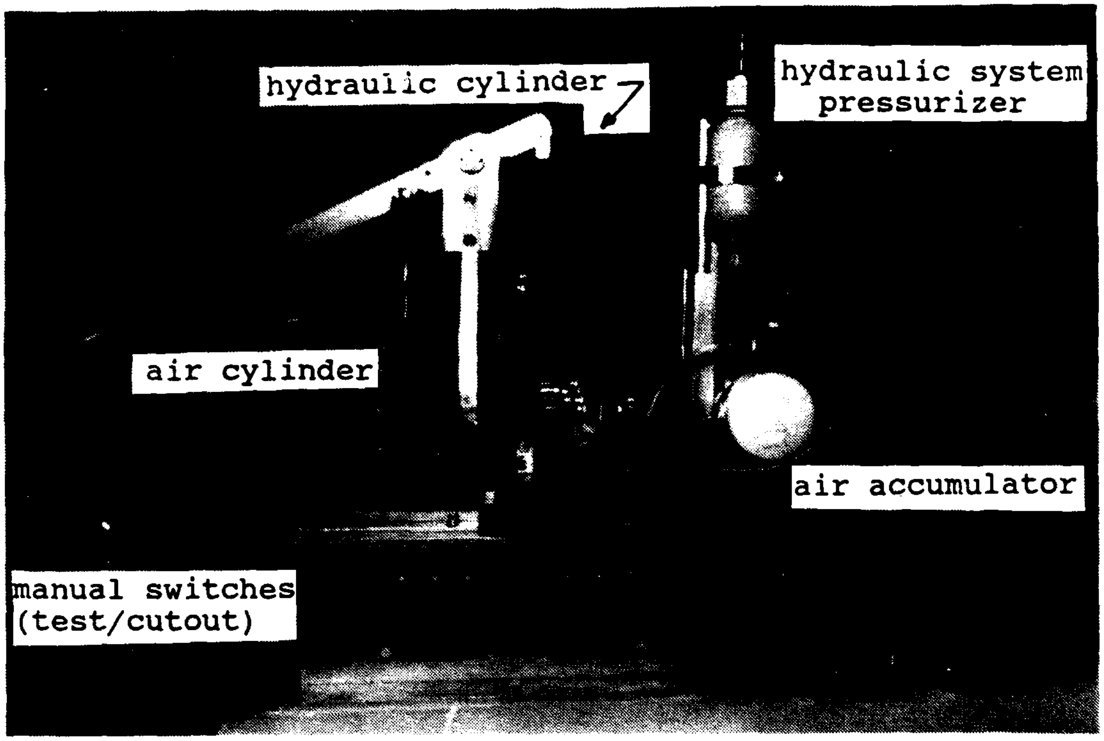


Figure 11. Novel Actuator Test Bed

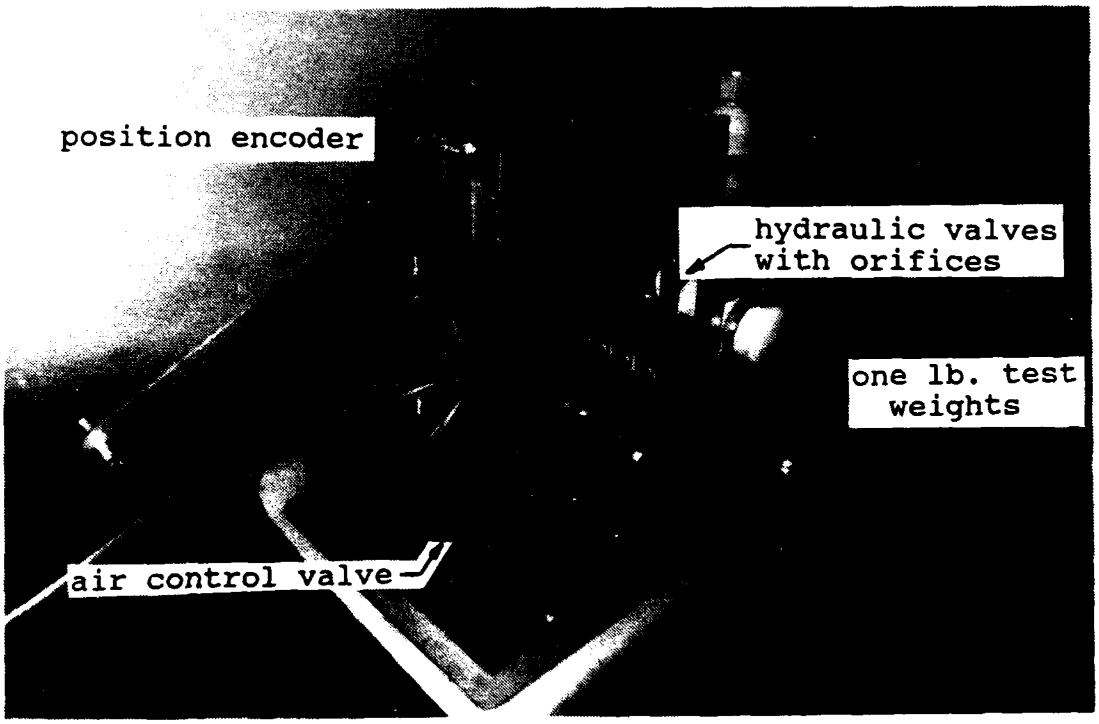


Figure 12. Novel Actuator Test Bed

orifice. Thus, by ordering different combinations of valves open, it was possible to control the flow from one side of the hydraulic cylinder to the other with a choice of from one to 15 times the area of the smallest orifice.

Because the hydraulic cylinder demonstrated a susceptibility to inducing air into the closed hydraulic system through its rod gland seals, a hydraulic system pressurizer was installed. This consisted of a fluid reservoir piped into the hydraulic system with valves to isolate it from the system (Figures 11 and 12). Prior to system operation, the reservoir was pressurized and then the isolation valves were opened. With the orifice solenoid valves also open, the pressure in both chambers of the cylinder equalized and forced the cup-type rod gland seals out against the rod and the seal housing, preventing air from being drawn in by piston movement. The pressurizer isolation valves were then closed and the system was ready for operation. The hydraulic system integrity was maintained, and the initial system pressure of between 30 and 40 psi was sustained throughout repeated operations without periodic repressurization.

B. EXPERIMENTAL SETUP

The arrangement used for taking data for model validation is shown in Figure 13. Two Kaman pressure transducers were installed at pressure taps on the hydraulic cylinder in order to measure the hydraulic pressure

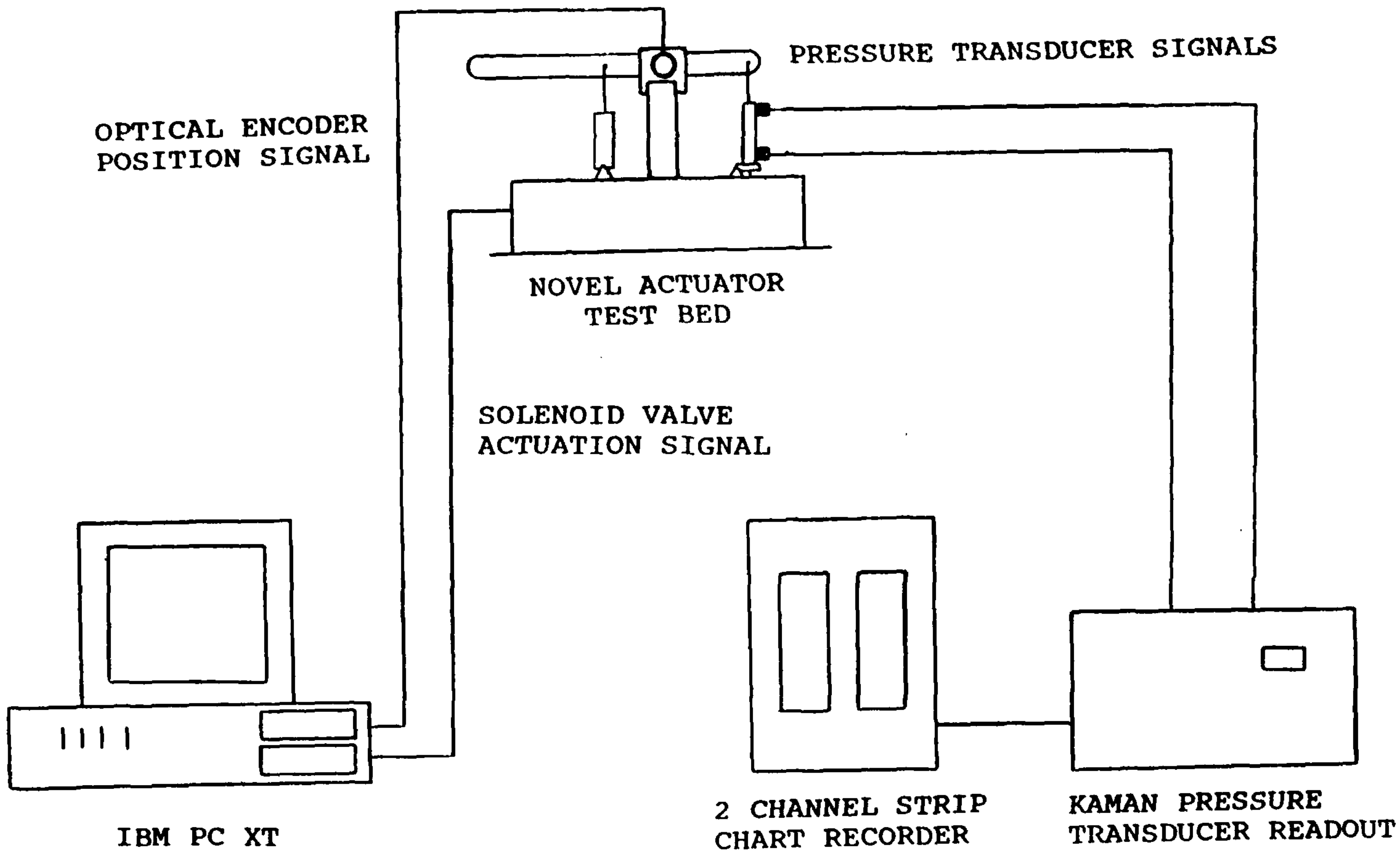


Figure 13. Experimental Setup

difference across the orifices (P_4 - P_5). The pressure difference signal output was recorded on a two channel strip chart recorder. An Omega series PX302 pressure transducer was placed on the supply air accumulator to monitor P_1. Calibration of the Kaman transducers was accomplished using a differential oil piston pressure gage calibrator, and the results are presented in Appendix C. Calibration of the Omega transducer (number P4) was accomplished in concurrent work by Verbos [Ref. 11]. Position and time data acquisition was done by the computer, using programs written to accomplish the tasks. All programs to test the operation of the Novel Actuator Test Bed, acquire data, and eventually implement automatic position control, were written in compiled BASIC for use with the Borland Turbo BASIC compiler [Refs. 12;13]. Programs related to data acquisition are shown in Appendix D. The TESTOP program was used to run the Novel Actuator Test Bed mechanism through its full range of motion for both lowering and raising moves, while recording encoder position in counts for a particular time in a data storage matrix. Time was measured by the PC system clock, accessible through a Turbo BASIC software command, with resolution in microseconds. Termination of a data run was effected by the operator at the finish of linkage travel through closing of a "kill switch" which was hard wired to an unused bit on the SEC-PC board input port. When the TESTOP program detected kill switch activation, the contents of the data storage matrix were written to a file, which was later manipulated using the MANIP program. This transformed the data from encoder counts to the different coordinates of interest and rendered output suitable for use with a computer graphing program. The program sample rate capability with the described computer configuration was approximately one encoder read every 4.66 milliseconds, or about 200 reads per second.

A set of five flat plate circular one pound weights suspended through their centers by lubricated bushings were used to provide different load conditions up to the design payload limit. Data collection runs were made for each orifice for full range of motion raising and lowering moves, and load conditions from zero to five pounds, in one pound increments.

C. RESULTS

Data presentations were made for the two extremes of loading conditions, no load and five pounds, and the intermediate load condition of two pounds. Hydraulic pressure difference strip chart recordings for each load condition, orifices one through four, raising and lowering motion, can be seen in Appendix E. Actual position versus time data for the same conditions, referenced to the hydraulic piston position coordinate X_1, are shown in graph format in Appendix F.

To obtain the best possible correlations between the theoretical predictions and actual data, all measurable quantities were recorded as accurately as possible and entered into the DSL simulation program for the pertinent motion case. Linkage bar dimensions were measured with a dial caliper accurate to 0.001 inch, and foundation lengths were measured with a machinist's scale to within 0.1 inch. Calculations for the mechanism's cumulative mass moment of inertia were made assuming that the load and piston masses were concentrated masses acting at their points of attachment on the linkage bar. The linkage bar was weighed and assumed to be a rectangular parallelopiped for moment of inertia calculation purposes. The hydraulic fluid density was measured using a hydrometer, and was corrected to ambient temperature. The orifice sizes were machined as accurately as possible on a lathe. the closest verification possible of orifice size was a measurement of the tool bit shank size with a micrometer, due to the extremely small sizes required.

A calculation for the orifice discharge coefficient was made for each load condition, effectively calibrating the orifices for each data run. Hydraulic piston speeds were calculated using change in position and change in time values picked from the straight line portions of the graphs in Appendix F. Then a value for volumetric flow was calculated using Equation (43). Equation (42) was solved for C_d giving:

$$C_d = \frac{Q_H}{A_{oh}\sqrt{\frac{2}{\rho}(P_4 - P_5)}} \tag{50}$$

A summary of the calculated C_d values is shown in Table 1.

TABLE 1

SUMMARY OF C_d VALUES

Orifice No.	Orifice Area	Motion	Load 0 lb.	Load 2 lb.	Load 5 lb.
1	.0001917	lower	.4894	.5081	.5775
1	.0001917	raise	.4928	.4348	.2513
2	.0003801	lower	.6365	.6726	.7891
2	.0003801	raise	.6540	.4203	.3803
3	.0007670	lower	.5678	.6333	.6469
3	.0007670	raise	.5761	.5714	.3531
4	.0014522	lower	.4355	.4251	.4699
4	.0014522	raise	.4084	.4000	.2654

The following graphical comparisons of actual versus predicted data are for the conditions representing the corners of the Novel Actuator Test Bed operating envelope. Figures 14, 15, 16, and 17 show no load and five pound load raising and lowering motion comparisons for the smallest and largest orifices. The closest agreement between model and hardware occurs in the case of lowering motions.

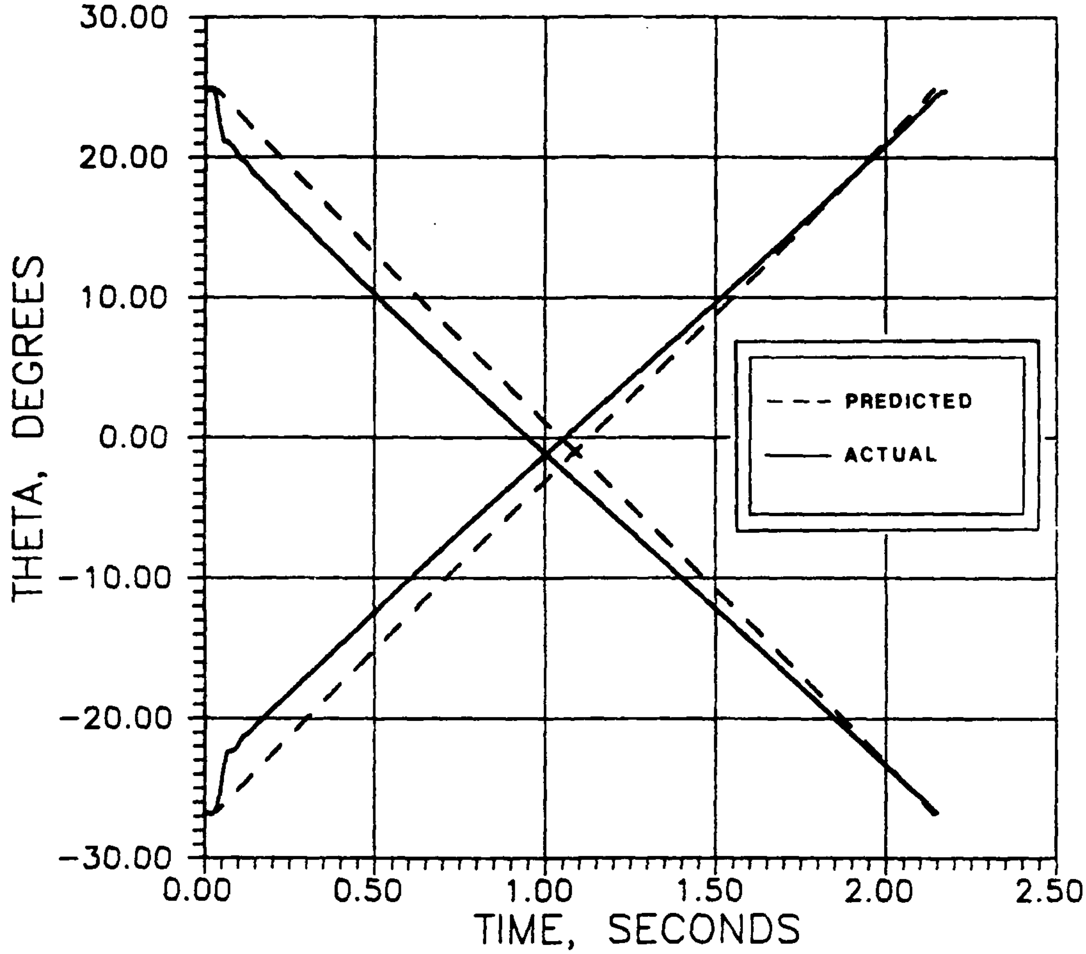


Figure 14. Orifice 1, No Load, Raising and Lowering Motions

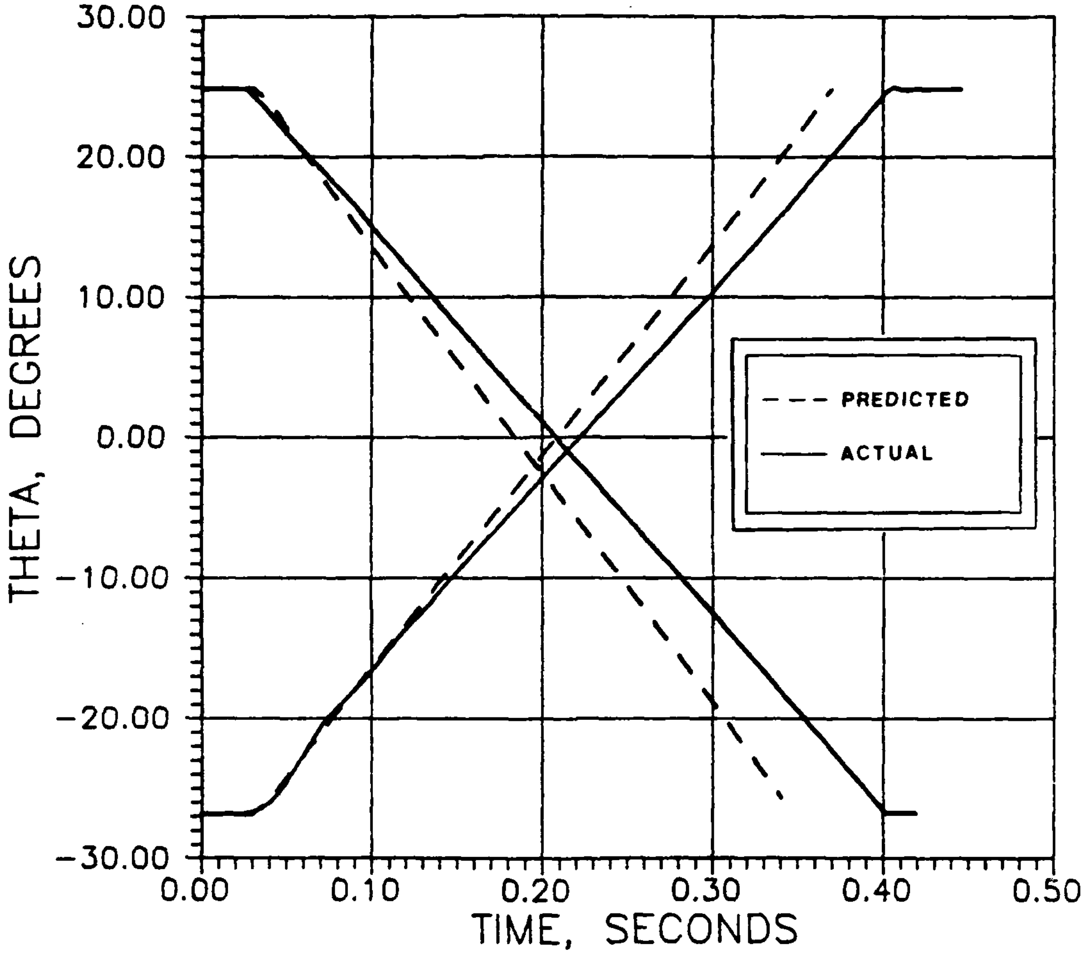


Figure 15. Orifice 4, No Load, Raising and Lowering Motions

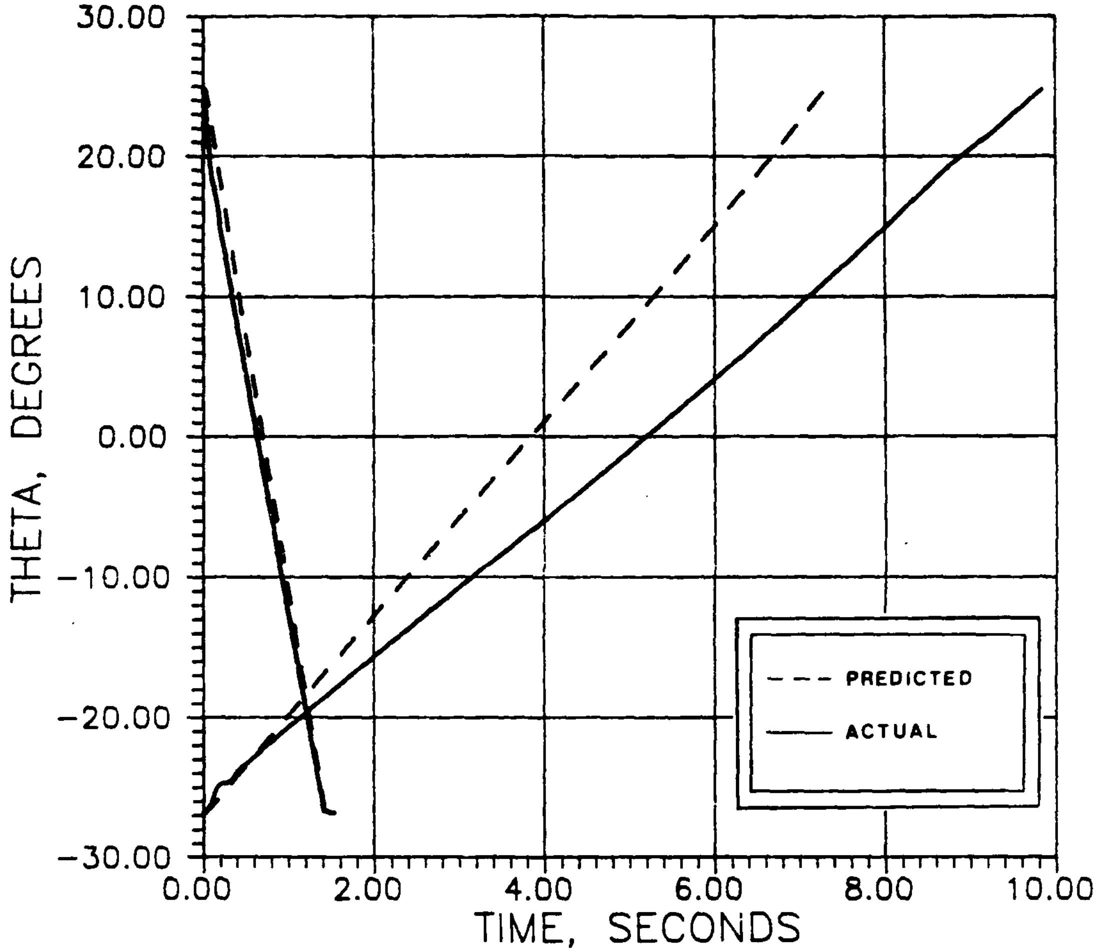


Figure 16. Orifice 1, 5 Lb Load, Raising and Lowering Motions

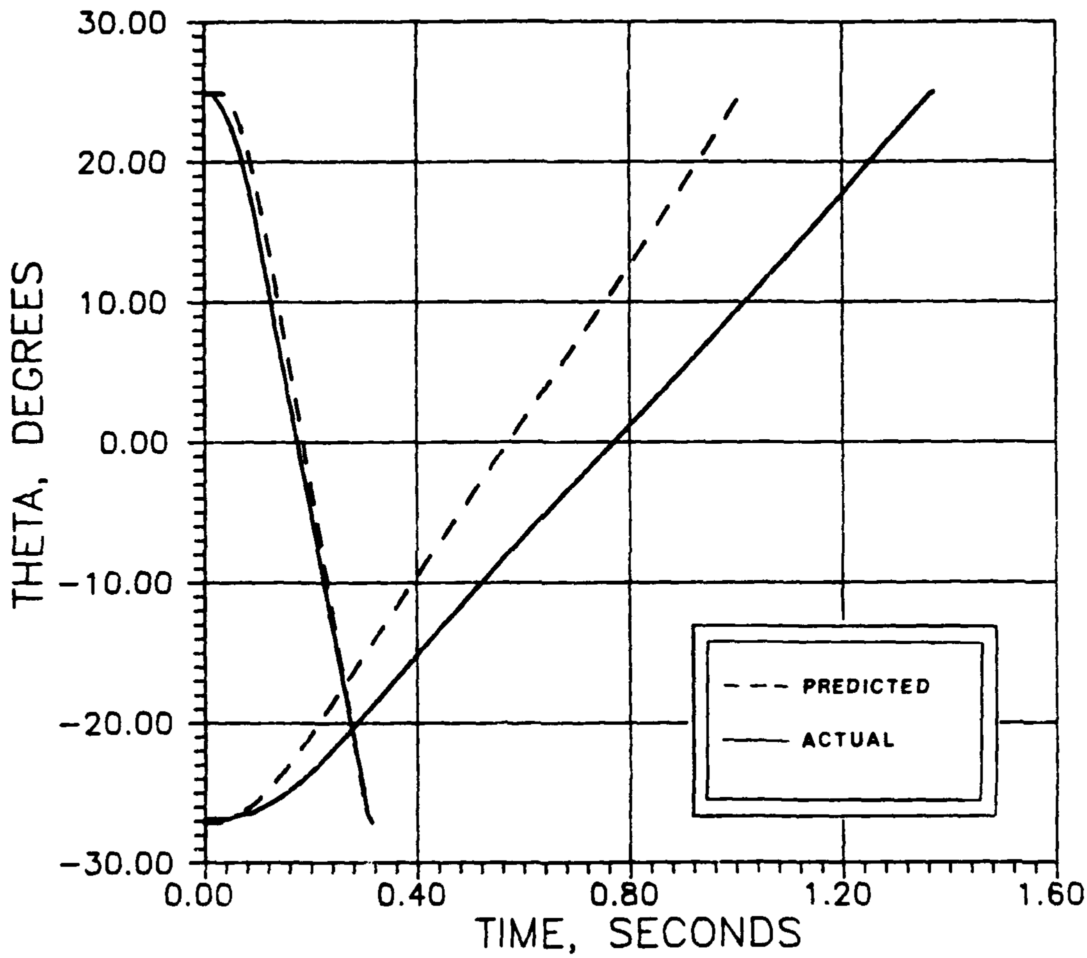


Figure 17. Orifice 4, 5 Lb Load, Raising and Lowering Motions

Additionally, the no load predictions were generally better than the loaded cases. It was anticipated that accounting for second order approximation effects, such as friction at the rotating joints and at the cylinder walls and glands, would have produced better model data. The predicted results for the most part fell within an experimental error factor of ten percent, so it was decided that the theoretical model offered an acceptably accurate description of system dynamics throughout the operating range since the purpose of the model was to develop a controller.

IV. CONTROL DESIGN

The design and implementation of an automatic control system was accomplished through use of the simulation program and the Test Bed hardware. The simulation model was used to assist in the choice of controller characteristics. A computer program using a discrete approximation to the controller was written to adapt the control scheme to the existing hardware. The actual system performance was then recorded and compared with the model predictions.

A. DESIGN METHODS

The objective of the control system design was to develop a controller capable of meeting the stated performance guidelines. The system needed to automatically position the Novel Actuator Test Bed mechanism, or "plant," as described by Equation (20) and the other supporting equations, in response to an ordered destination different from its current position. Figure 18 illustrates the type of control system required. The plant was inherently nonlinear, chiefly because of the hydraulic cylinder force term which had dependence on the nonlinear turbulent orifice flow equation. Since the actual control method of four distinct fixed area orifices in various combinations was being used to roughly approximate servo control, it was not practical to try the linearization schemes possible when

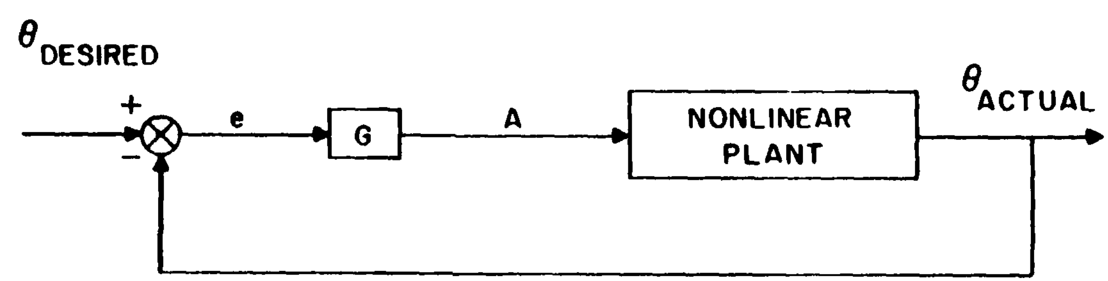


Figure 18. General Position Control System

using a variable opening device such as a servo valve [Ref. 7]. Consequently, classical frequency domain control design methods were not applicable in this case. However, from exercising of the DSL simulation, it was realized that the system was very stable. With the availability of an extremely high sample and order delivery rate by the computer and interface board, it was decided to use a proportional type controller, and select the gain for the controller by a cut and try process of matching the gains to the desired response characteristics.

The controller was developed using "worst case" design principles; that is, the design requirements were to be met or exceeded while the mechanism was operating at its maximum limits. In this instance, "worst case" meant meeting the response time criteria while safely raising the maximum five pound load. The DSL simulation program that was used to predict system response was also used to investigate likely candidates for the proportional gain G. The control loop was closed in the program by calculating an error signal during each integration step equal to:

$$e = \theta_{DESIRED} - \theta_{ACTUAL} \tag{51}$$

The hydraulic control orifice area was then programmed as:

$$A_{oh} = G\ e \tag{52}$$

The program was run several times for different parameter values of G, so that the resultant response curves bracketed desired performance. Figure 19 shows the response curves produced for a range of five values of G. Based on this output, a value of G equal to 0.002 was seen to deliver acceptable response as delineated by the design criteria.

B. CONTROL SCHEME IMPLEMENTATION

Figure 20 shows the ideal controller gain curve and the discrete approximation to the curve that was implemented on the Novel Actuator Test Bed. A four point approximation to the curve was chosen because repeated state changes of the valves during motion was considered less desireable from a reliability standpoint. Also, effective areas greater than the largest orifice size (eight times that of the smallest orifice) corresponded to a larger error value than the full range of motion of the mechanism. This was a promising indication that several levels of performance improvement over the chosen design criteria were possible without any hardware alteration. An automatic control program called CONTROL was written and is listed in Appendix G. The error values were translated from degrees to encoder counts, and orifice control valve switching was effected by logic statement tests within the program. The smallest acceptable error was chosen to be when the mechanism was approximately .3 degrees away from ordered position. A block diagram of

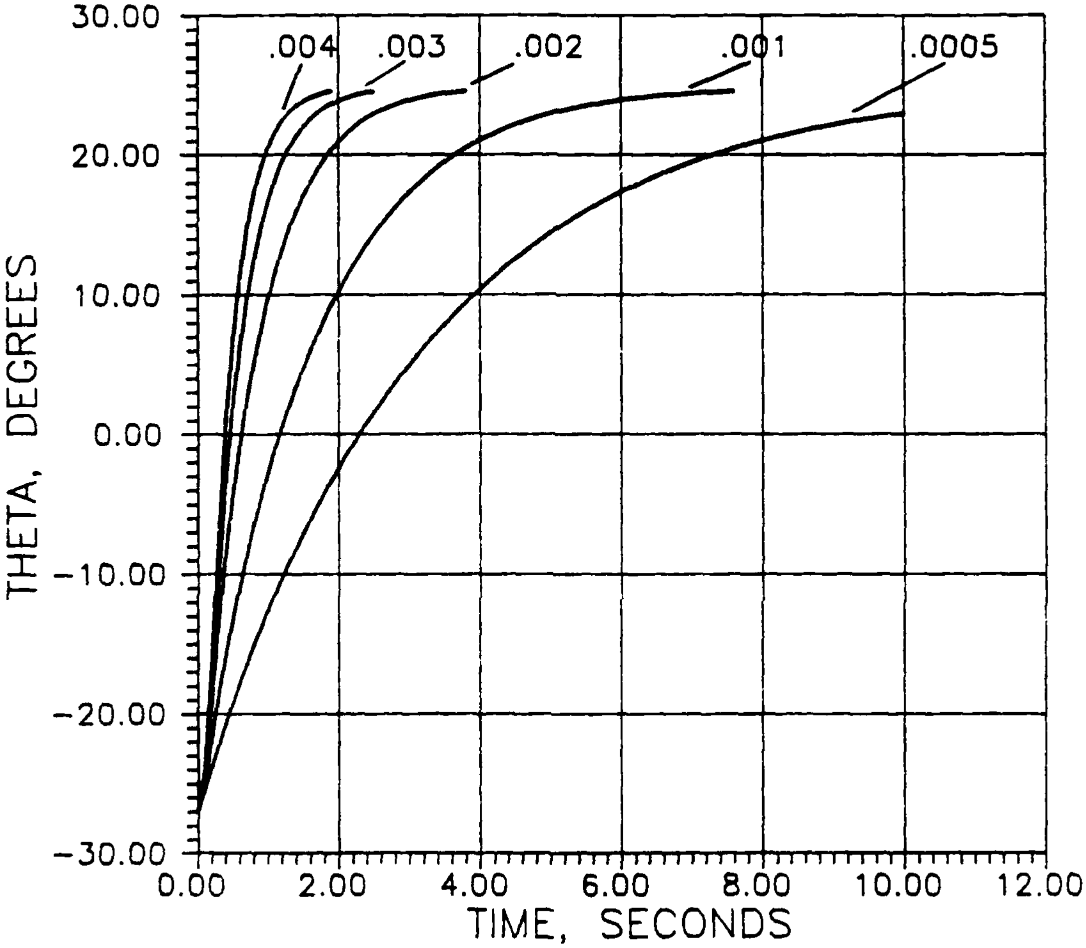


Figure 19. Response Predictions for Various Gains, G

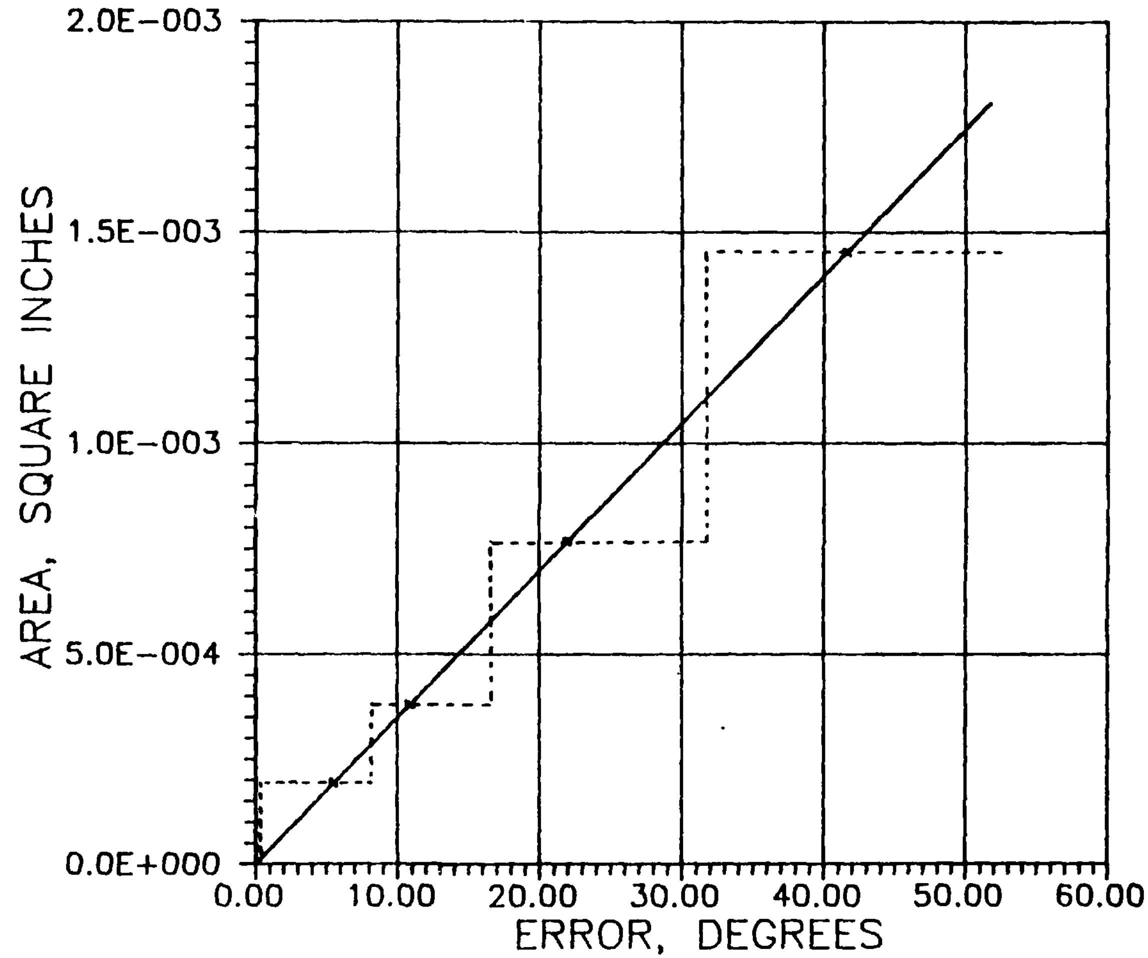


Figure 20. Continuous Controller Gain Curve and Discrete Step Approximation

the specific control system showing the hardware and software relationships is illustrated in Figure 21.

Figure 22 shows the predicted and actual system response curves for the raising of a five pound load throughout the entire range of motion. The actual response using a discretized approximation to the chosen gain proved to be very close to the predicted response using a continuous gain curve, and all the design guidelines were successfully met. Positioning accuracy was less than +/- .1 inch, which was half as small as the allowable limit. The switching of the orifice valves can be discerned on the actual response curve as points of changing slope. The controller was rigorously tested for all motions and load conditions within the operating envelope, and proved to be both accurate and robust.

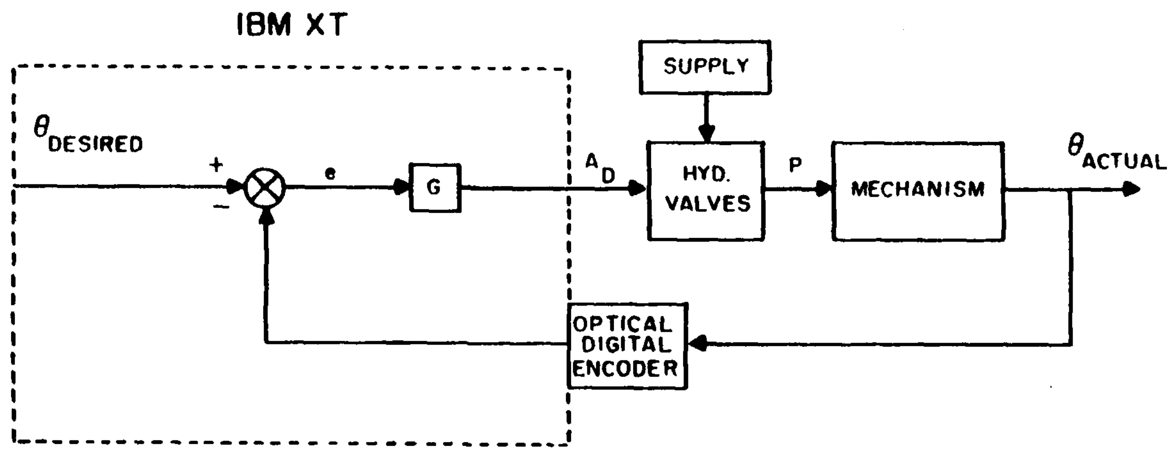


Figure 21. Specific Position Control System

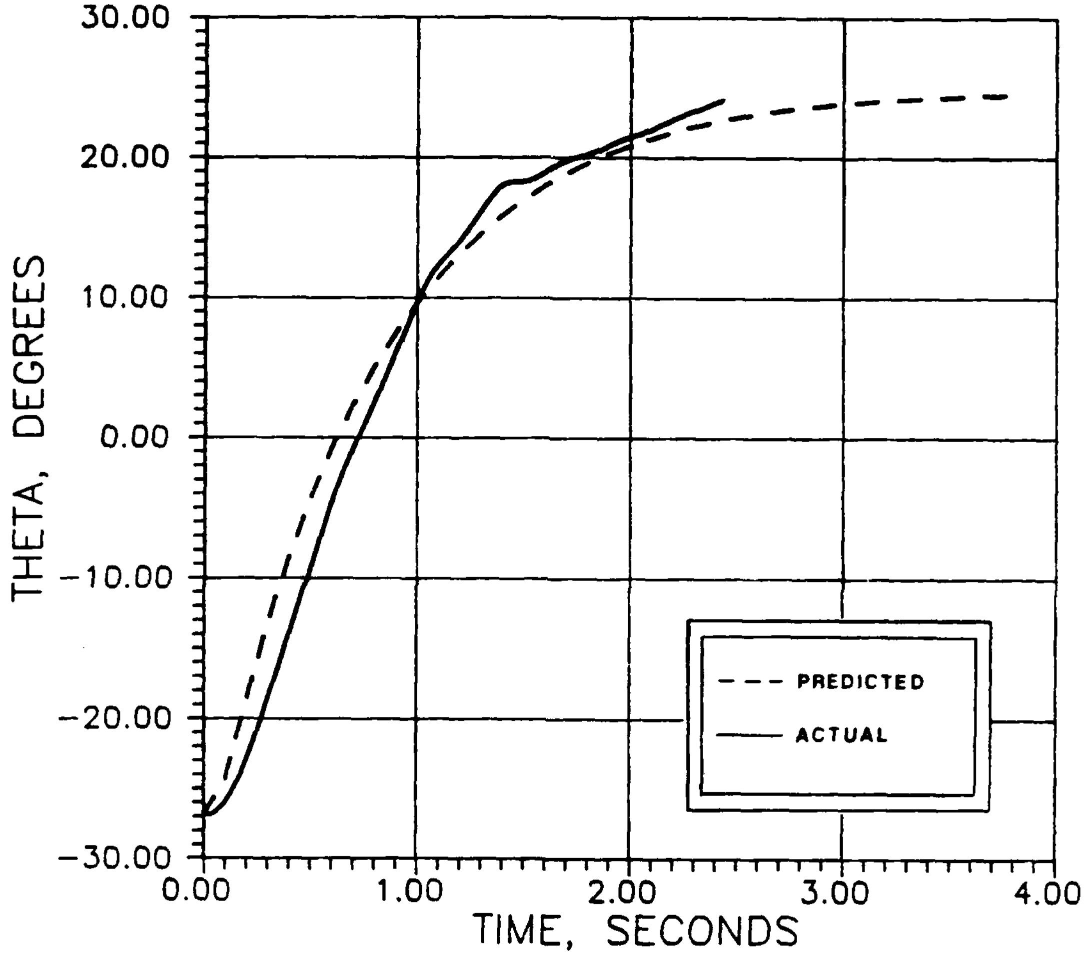


Figure 22. Test Bed Response to a Full Range of Motion Order with Maximum Load

V. CONCLUSIONS AND RECOMMENDATIONS

All major objectives sought in the investigation of the fitness of the Novel Actuator for robotic application on a mobile platform were successfully achieved. A suitable system model was found which acceptably described dynamic behavior, and was highly effective in use as a design tool for creating an automatic control system. Although the theoretical plant model was nonlinear in nature, the stability of the system enabled the cut and try method to be effective in controller design.

The Novel Actuator Test Bed offered outstanding return on minimal investment. It was an excellent forum for proof of theory, and implementation of the control design in hardware. The use of the digital optical encoder to sense position eliminated the need for analog-to-digital data conversion, and the SEC-PC interface board's multiple capabilities contributed to system simplicity. The IBM PC was well suited as a controlling apparatus, and along with the compiled BASIC software operating environment it helped provide a well behaved and rugged system.

The automatic controller easily conformed to the specified design guidelines, and sufficient overhead exists in installed hardware to make significant improvements in performance.

Observations made during mechanism operations have suggested the following design recommendations for application to proposed Novel Actuator detail design and development [Ref. 5]. Recommendations are segregated into general areas of applicability, and an intuitive determination of whether implementation would be a high cost (HC), low cost (LC), or no cost (NC) improvement has been made.

A. HYDRAULIC SUBSYSTEM RECOMMENDATIONS

The primary design problem for this actuator is entrapment and removal of air from the closed hydraulic system. Bleed connections were established at system high points, but great difficulty was experienced in removing all trapped air because of the dead ends at the cylinder chambers. Trapped air manifested itself as a small region within which positioning could not be accomplished on initial start of motion. Compression of the trapped air was detected as a slight oscillation or spring effect at the start of motion, as seen in the graphs in Appendices E and F. Its presence also detracted from overall system stiffness, as discussed in chapter two. Air must be prevented from entering the closed system, and a simple and consistent provision should be made for its removal, should it enter the system. Improvements which could help this problem are:

1. Switch from a petroleum based hydraulic fluid to a water based or water-glycol hydraulic fluid, which may have a lesser tendency to entrain air bubbles (LC).

2. Install bleed connections at each end of the hydraulic cylinder to allow bleeding of the individual cylinder chambers (LC).

3. Obtain a cylinder with more effective or multiple rod gland seals to prevent air from entering past this joint (HC).

The three port solenoid valves controlling access to the hydraulic orifices limit the driving force air supply pressure, as installed in their present configuration. When the air cylinder is energized in the raising direction with the hydraulic valves closed, a point can be reached where the valves are driven off their seats (from fluid pressure acting beneath the seat) causing the mechanism to move. By increasing the air supply pressure, the force on the underside of the valve is eventually able to overcome the force of the internal valve springs, and the valves cannot stop fluid flow. The system requires that the control valves for the orifices have the characteristics of positive shut off in both directions with zero leak through. Two solutions to this limitation are:

1. Install another solenoid valve between chamber 4 of the hydraulic cylinder and the parallel arrangement of control valves, oriented so that it would open against pressure when a raising motion is ordered (LC). Minor software changes to test bed operating programs would be required to open this valve during every movement of the mechanism (NC).

2. Investigate and procure other valve types capable of meeting the positive shut off, no leak through criteria (HC).

The Kaman pressure transducer measuring P_5 was subjected to over-range pressures when testing the higher load conditions, as shown by the five pound load, lowering motion, P_5 strip chart included in Appendix E (worst case from a pressure standpoint). The manufacturer's guarantee was to 200 psi and the average transient for the case shown was 300 psi. The subject transducer was calibrated up to this high pressure, and found to be essentially linear, however, no provision for pressure measurement was available in the event that a higher air supply pressure was desired in order to increase the load capacity of the system. If hydraulic pressure measurements are necessary in a next generation actuator, it is recommended that 1000 psi transducers be installed to allow flexibility in the choice of supply pressure (HC).

B. PNEUMATIC SUBSYSTEM RECOMMENDATIONS

The five port, three position control valve performed well in the control of the air cylinder. The only reason to change from this configuration would be if another choice is more suitable from the standpoint of reliability. Because of its essentially "on or off" nature, the system as a whole was subjected to the full air supply pressure immediately after air valve activation. This tended to magnify the spring-like effects at the start of motion due to entrained air in the hydraulic system, and possible linkage and foundation deflections, when subject to the full force translation of the maximum supply pressure. Investigations could be conducted on the merits of using a proportional servo valve vice the present control valve (HC). Major tradeoff issues would be possible increases in component life and smoothness of operation versus added complexity of the control system and additional computer interface required for conversions between analog and digital data.

C. CONTROL SUBSYSTEM RECOMMENDATIONS

A simple four point discretization to the chosen gain curve was implemented and performed well, with no perceptible decrease in the sample and order rate due to the logic testing statements. It is recommended that a series of controllers be designed to take advantage of more of the available discrete points and document available performance levels with the existing hardware (NC).

As described in Kuo [Ref. 14], speed information can be extracted from the encoder data by a simple calculation. Addition of speed information feedback to the automatic control system should be investigated for its performance improvement potential (NC).

Data from the pressure transducers offers a means of load sensing which can be used to tailor the actions of the controller. This could also be accomplished through use of strain gages placed on the linkage at strategic points. The sophistication of the control scheme could be increased through load sensing at the cost of more interface hardware (HC).

D. STRUCTURAL IMPROVEMENT RECOMMENDATIONS

The Novel Actuator Test Bed was a low precision machine built to test concepts and designs. Several structural modifications could improve the accuracy and reliability of the mechanism:

1. Use bearings on all rotating joints, cylinder pins, and clevises (HC).
2. Devise a better mount for the optical encoder to fix the body and prevent cable wear (LC).
3. Statically balance the linkage arm to eliminate unnecessary mechanism loading (LC).

As stated in Reference 15, linear actuators acting on rotary joints are less desireable because the effective power transmission ratio varies with joint position. An actuator with a configuration similar to the conceptual schematic of Figure 1 would rectify this deficiency (HC).

E. SCOPE OF PROJECT RECOMMENDATIONS

The results of the Novel Actuator design project have proven that the concept has a great deal of promise for application to robotic devices. A logical extension of this project is to implement the actuator on a manipulator with two or more degrees of freedom and address the control issues that arise from that application.

Future end users of such a robotic device need to assist in the clarification of a more specific design goal. A trial task should be developed, and mobile base dynamics need to be included in the task specification so that their effects on the system may be dealt with.

APPENDIX A

THEORETICAL MODEL SIMULATION PROGRAMS

```
TITLE NOVEL ACTUATOR DYNAMICS - RAISING MOVEMENT
INITIAL
D        REAL*8 P1,AP,AH,L11,L12,L13,W,THDOT0,TH0,M,J
D        REAL*8 TERM1,TERM2,DELPH,AOH,PI,THDDOT,THDOT
D        REAL*8 TH,LFP,LFH,LP,LH,BP,BH,ERR,WY,FPY,FHY
D        REAL*8 FP,FH,PHIP,PHIH,TM1,TM2,X1,X2,X1DOT,HCYL
D        REAL*8 PCYL,ETAP,ETAH,ALPHAP,ALPHAH,G,THDES,RHO
D        REAL*8 CD,WLY,L14,WL
CONST    P1 = 50.0,       AP = 0.8866,     AH = 0.1994
*           PSI              IN**2            IN**2
*
CONST    L11 = 2.6235,    L12 = 2.635,     L13 = 14.980
*            IN.              IN.              IN.
*
CONST    W = 5.0,         THDOT0 = 0.0,     TH0 = -0.4693981
*          LB.              RAD/S              RAD
*
CONST    LFP = 10.3,      LFH = 10.6,       LP = 8.643
*          IN.              IN.                IN.
*
CONST    LH = 11.46,      L14 = 6.402,      WL = 0.650
*          IN.              IN.                LB.
*
CONST    G = 0.004,       THDES = .4341166
*           ND               RAD
*
CONST    RHO = 0.0000808
*        LB-SEC**2/IN**4
*
         M = W
*
         J = .1838 + (.5823 * M)
*               IN-LB-S**2
*
         PI = 4.0 * ATAN(1.0)
*
         BP = ACOS(L12/LFP)
*
         BH = ACOS(L11/LFH)
*
         FP = AP *P1
*
         CD = .2513
*        CD = .3803
```

```
*       CD = .3531
*       CD = .2654

DERIVATIVE
NOSORT
        IF (AOH .EQ. 0.0) GO TO 10

SORT
        WY = W * COS(TH)
        WLY = WL * COS(TH)
        FPY = FP * SIN(ALPHAP)
        FHY = FH * SIN(ALPHAH)
        FH = AH * DELPH
        PHIP = BP + TH
        PHIH = BH - TH
        TM1 = LFH**2 + L11**2 - 2*LFH*L11*COS(PHIH)
        TM2 = LFP**2 + L12**2 - 2*LFP*L12*COS(PHIP)
        X1 = LH - TM1**0.5
        X2 = TM2**0.5 - LP
        X1DOT = (LFH*L11*THDOT*SIN(PHIH))*TM1**(-0.5)
        HCYL = LH - X1
        PCYL = LP + X2
        ETAP = ASIN((LFP/PCYL) * SIN(PHIP))
        ALPHAP = PI - ETAP
        ETAH = ASIN((LFH/HCYL) * SIN(PHIH))
        ALPHAH = PI - ETAH
        DELPH = (AH*X1DOT/(CD*AOH*(2/RHO)**.5))**2
        THDDOT = (1.0/J)*(FPY*L12 - FHY*L11 - WY*L13 -
&WLY*(L14+L12))
        THDOT = INTGRL(THDOT0,THDDOT)
NOSORT
        GO TO 20
10      THDDOT = 0.0
        THDOT = 0.0
        DELPH = 0.0
SORT
20      TH = INTGRL(TH0,THDOT)
        T2DDEG = THDDOT * (180.0/PI)
        T1DDEG = THDOT * (180.0/PI)
        TDEG = TH * (180.0/PI)
DYNAMIC
        ERR = THDES - TH
        AOH = G * ERR
*       AOH = .0001917 * STEP(.030)
*       AOH = .0003801 * STEP(.030)
*       AOH = .000767 * STEP(.030)
*       AOH = .0014522 * STEP(.030)
*       IF (TH .GE. 0.4341166) CALL ENDRUN
        IF (ERR .LE. .005) CALL ENDRUN
TERMINAL
CONTRL FINTIM = 10.0,DELT = .005
SAVE .01,T2DDEG,T1DDEG,TDEG,DELPH,AOH,X1DOT,X1,X2
PRINT .1,TDEG,T1DDEG,X1,X1DOT
```

```
END
PARAM    G = 0.003
END
PARAM    G = 0.002
END
PARAM    G = 0.001
END
PARAM    G = 0.0005
END
GRAPH(DE=TEK618) TIME,TDEG
GRAPH(DE=TEK618) TIME,DELPH
END
STOP
```

```
TITLE NOVEL ACTUATOR DYNAMICS - LOWERING MOVEMENT
INITIAL
D        REAL*8 P1,AP,AH,L11,L12,L13,W,THDOTO,THO,M,J
D        REAL*8 TERM1,TERM2,DELPH,AOH,PI,THDDOT,THDOT
D        REAL*8 TH,LFP,LFH,LP,LH,BP,BH,WY,FPY,FHY,FP
D        REAL*8 FH,PHIP,PHIH,TM1,TM2,X1,X2,X1DOT,HCYL
D        REAL*8 PCYL,ETAP,ETAH,ALPHAP,ALPHAH,G,THDES
D        REAL*8 RHO,CD,WLY,L14,WL
CONST    P1 = 50.0,        AP = 0.8099,      AH = 0.1994
*           PSI                 IN**2             IN**2
*
CONST    L11 = 2.6235,     L12 = 2.635,      L13 = 14.980
*            IN.                IN.               IN.
*
CONST    W = 2.0,          THDOTO = 0.0,      THO = 0.4341166
*          LB.               RAD/S                RAD
*
CONST    LFP = 10.3,       LFH = 10.6,        LP = 8.643
*          IN.               IN.                  IN.
*
CONST    LH = 11.46,       L14 = 6.402,       WL = 0.650
*          IN.               IN.                  LB.
*
CONST    G = .006,         THDES = -.4693981
*          ND                  RAD
*
CONST    RHO = 0.0000808
*        LB-SEC**2/IN**4
*
         M = W
*
         J = .1838 + (.5823 * M)
*               IN-LB-S**2
*
         PI = 4.0 * ATAN(1.0)
*
         BP = ACOS(L12/LFP)
*
         BH = ACOS(L11/LFH)
*
         FP = AP *P1
*
*        CD = .5081
*        CD = .6726
*        CD = .6333
         CD = .4251
*
DERIVATIVE
NOSORT
         IF (AOH .EQ. 0.0) GO TO 10
SORT
         WY = W * COS(TH)
         WLY = WL * COS(TH)
```

```
         FPY = FP * SIN(ALPHAP)
         FHY = FH * SIN(ALPHAH)
         FH = AH * DELPH
         PHIP = BP + TH
         PHIH = BH - TH
         TM1 = LFH**2 + L11**2 - 2*LFH*L11*COS(PHIH)
         TM2 = LFP**2 + L12**2 - 2*LFP*L12*COS(PHIP)
         X1 = LH - TM1**0.5
         X2 = TM2**0.5 - LP
         X1DOT = (LFH*L11*THDOT*SIN(PHIH))*TM1**(-0.5)
         HCYL = LH - X1
         PCYL = LP + X2
         ETAP = ASIN((LFP/PCYL) * SIN(PHIP))
         ALPHAP = PI - ETAP
         ETAH = ASIN((LFH/HCYL) * SIN(PHIH))
         ALPHAH = PI - ETAH
         DELPH = (AH*X1DOT/(CD*AOH*(2/RHO)**.5))**2
         THDDOT = (1.0/J)*(FHY*L11 - FPY*L12 - WY*L13 -
        &WLY*(L14+L12))
         THDOT = INTGRL(THDOT0,THDDOT)
NOSORT
         GO TO 20
10       THDDOT = 0.0
         THDOT = 0.0
         DELPH = 0.0
SORT
20       TH = INTGRL(TH0,THDOT)
         T2DDEG = THDDOT * (180.0/PI)
         T1DDEG = THDOT * (180.0/PI)
         TDEG = TH * (180.0/PI)
DYNAMIC
*        AOH = G * (TH - THDES)
*        AOH = .0001917 * STEP(.030)
*        AOH = .0003801 * STEP(.030)
*        AOH = .000767 * STEP(.030)
         AOH = .0014522 * STEP(.030)
         IF (TH .LE. -0.4693981) CALL ENDRUN
TERMINAL
CONTRL FINTIM = 10.0,DELT = .0001
SAVE .001,T2DDEG,T1DDEG,TDEG,DELPH,AOH,X1DOT,X1,X2
PRINT .01,TDEG,T1DDEG,X1,X1DOT
*END
*PARAM    G = 0.005
*END
*PARAM    G = 0.004
*END
*PARAM    G = 0.003
*END
*PARAM    G = 0.002
*END
*PARAM    G = 0.001
*END
*PARAM    G = 0.0005
```

```
*END
*GRAPH(DE = TEK618) TIME,T1DDEG,TDEG
*GRAPH(DE = TEK618) TIME,DELPH
END
STOP
```

APPENDIX B

SOLENOID VALVE COMPUTER INTERFACE

A schematic diagram of the electrical connections and components used to provide an interface between the IBM PC and the pneumatic and hydraulic solenoid valves is shown in Figure B-1.

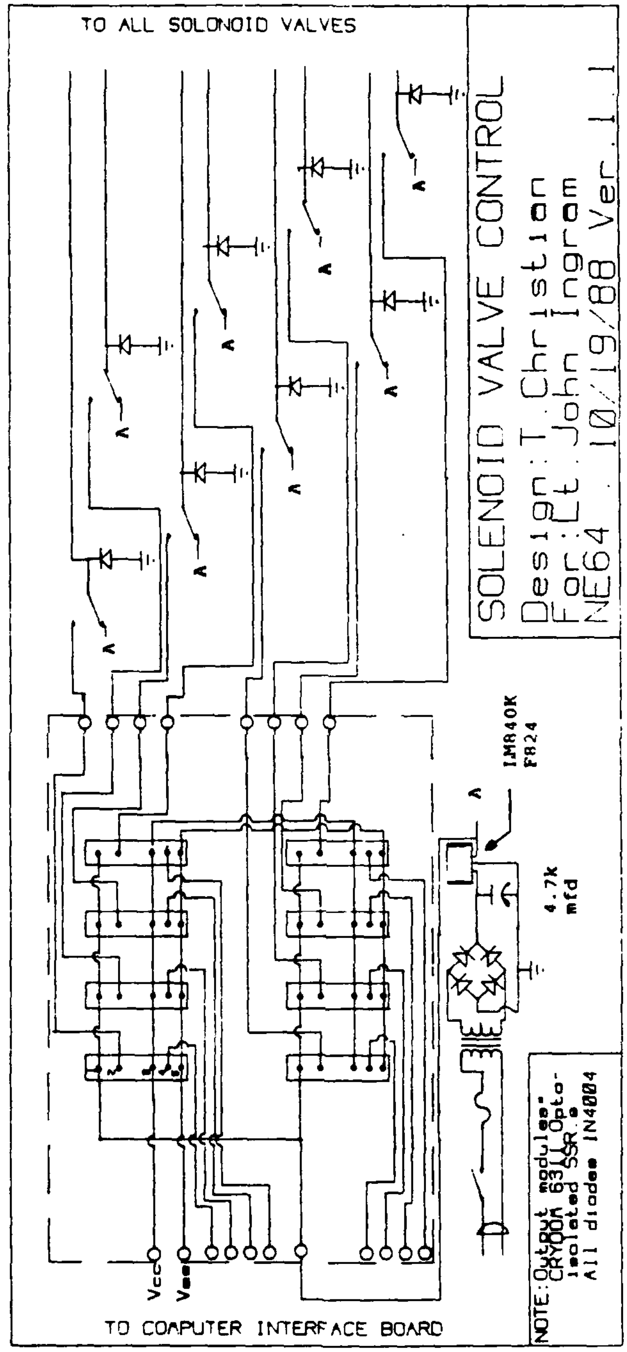


Figure B-1 Valve/Computer Interface

APPENDIX C

HYDRAULIC PRESSURE TRANSDUCER CALIBRATION DATA

Results of the calibration of the two channel Kaman pressure transducers are presented in the following table and graphically in Figure C-1. Channel A was used to measure P_5 and channel B measured P_4.

TABLE C-1

PRESSURE TRANSDUCER DATA

Reference Standard Pressure	Channel A Pressure	Channel B Pressure
0.0	0.0	-0.35
10.0	9.9	9.6
20.0	19.8	19.4
30.0	29.8	29.4
40.0	39.7	39.3
50.0	49.7	49.3
60.0	59.6	59.2
70.0	69.8	69.4
80.0	79.7	79.4
90.0	89.8	89.5
100.0	100.0	99.8
110.0	110.2	109.9
120.0	120.3	120.1
130.0	130.4	130.3
140.0	140.6	140.4
150.0	150.8	150.6
160.0	160.9	160.9
170.0	171.0	171.0
180.0	181.0	181.0
200.0	203.0	
225.0	227.0	
250.0	253.0	
275.0	278.0	
300.0	303.0	

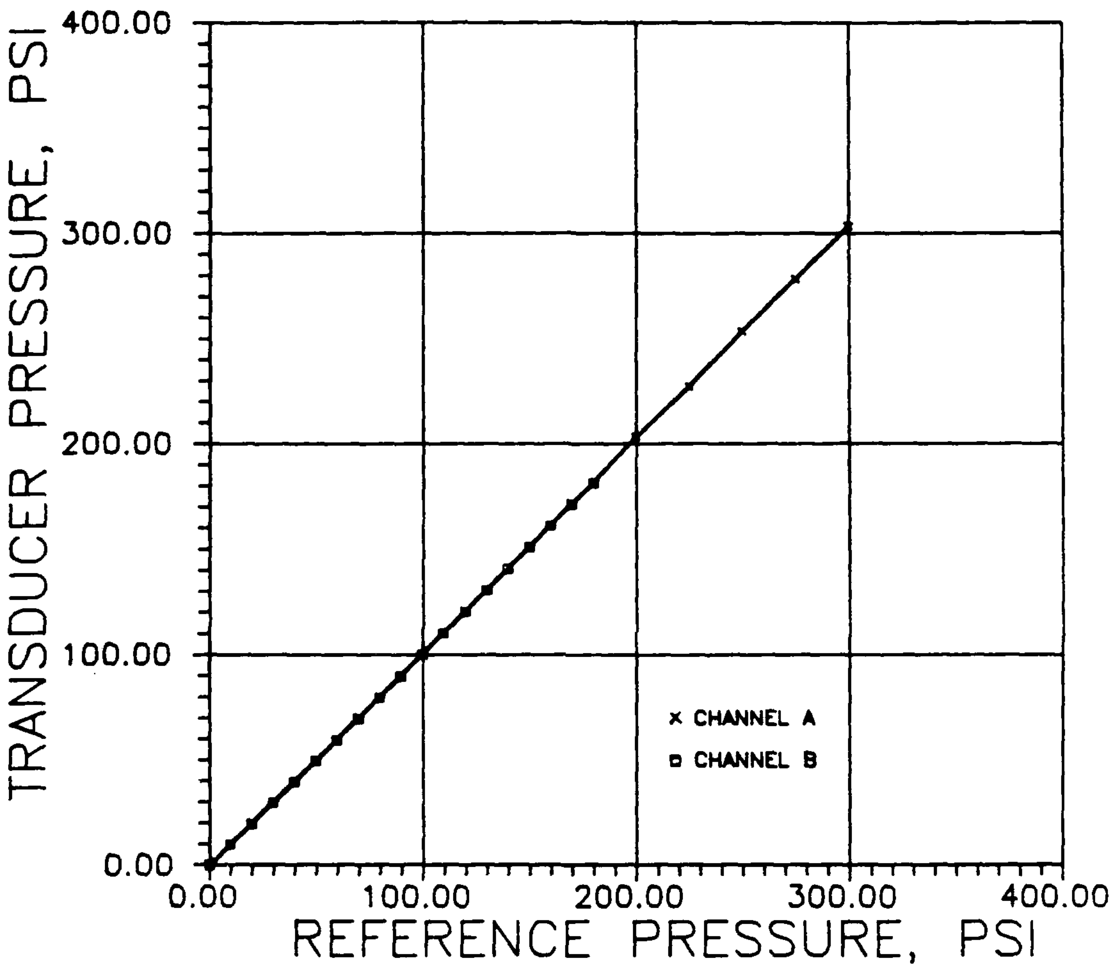


Figure C-1 Hydraulic Pressure Transducer Data

APPENDIX D

DATA ACQUISITION PROGRAMS

```
'                          TESTOP.BAS
'
' THIS PROGRAM WAS CREATED TO TEST THE OPERATION OF THE
' NOVEL ACTUATOR TEST BED, AND GATHER DATA FOR MODEL
' VERIFICATION.  WRITTEN 10-27-88
'                                              JDI.
'
'
'             LOCATE INTERFACE BOARD PORTS
'
C = &H330 : P = C + 3 : SOLENOIDS = C + 1
'
'             CLEAR COUNTERS
'
OUT C,0
'
'             SET VALVE PORT TO CLOSE ALL SOLENOIDS
'
OUT SOLENOIDS,255
'
'             DIMENSION ARRAY TO STORE TIME AND ENCODER DATA
'
OPTION BASE 1
DIM POSIDAT(2500,2)
'
'             DETERMINE TYPE OF MOVE
'
100 PRINT "ENTER 1 FOR RAISE, 2 FOR LOWER, 0 TO HALT
PROGRAM"
INPUT AORDER%
CLS
IF AORDER% = 0 THEN GOTO 400
AV% = AORDER%
'
'             SPEED SELECTION - CREATE HYDRAULIC VALVE TABLE
'
HV1% = 32 : HV2% = 16
HV3% = 8  : HV4% = 4
'
'             SELECT ORIFICES TO OPEN
'
200 PRINT "ENTER A FOUR DIGIT NUMBER CONSISTING OF"
    PRINT "1'S AND 0'S, 1 FOR OPEN AND 0 FOR CLOSED,"
    PRINT "FOR VALVES 1 THROUGH 4 CONSECUTIVELY."
INPUT HORDER%
```

```
IF HORDER% = 0001 THEN
    HV% = HV1%
    PRINT "OPENING ORIFICE 1"
ELSEIF HORDER% = 0010 THEN
    HV% = HV2%
    PRINT "OPENING ORIFICE 2"
ELSEIF HORDER% = 0011 THEN
    HV% = HV1% OR HV2%
    PRINT "OPENING ORIFICES 1 AND 2"
ELSEIF HORDER% = 0100 THEN
    HV% = HV3%
    PRINT "OPENING ORIFICE 3"
ELSEIF HORDER% = 0101 THEN
    HV% = HV3% OR HV1%
    PRINT "OPENING ORIFICES 1 AND 3"
ELSEIF HORDER% = 0110 THEN
    HV% = HV3% OR HV2%
    PRINT "OPENING ORIFICES 2 AND 3"
ELSEIF HORDER% = 0111 THEN
    HV% = HV3% OR HV2% OR HV1%
    PRINT "OPENING ORIFICES 1, 2, AND 3"
ELSEIF HORDER% = 1000 THEN
    HV% = HV4%
    PRINT "OPENING ORIFICE 4"
ELSEIF HORDER% = 1001 THEN
    HV% = HV4% OR HV1%
    PRINT "OPENING ORIFICES 1 AND 4"
ELSEIF HORDER% = 1010 THEN
    HV% = HV4% OR HV2%
    PRINT "OPENING ORIFICES 2 AND 4"
ELSEIF HORDER% = 1011 THEN
    HV% = HV4% OR HV2% OR HV1%
    PRINT "OPENING ORIFICES 1, 2, AND 4"
ELSEIF HORDER% = 1100 THEN
    HV% = HV4% OR HV3%
    PRINT "OPENING ORIFICES 3 AND 4"
ELSEIF HORDER% = 1101 THEN
    HV% = HV4% OR HV3% OR HV1%
    PRINT "OPENING ORIFICES 1, 3, AND 4"
ELSEIF HORDER% = 1110 THEN
    HV% = HV4% OR HV3% OR HV2%
    PRINT "OPENING ORIFICES 2, 3, AND 4"
ELSEIF HORDER% = 1111 THEN
    HV% = HV4% OR HV3% OR HV2% OR HV1%
    PRINT "OPENING ORIFICES 1, 2, 3, AND 4"
ELSE
    GOTO 200
END IF
PRINT
PRINT "STOP MOVE BY PRESSING KILL SWITCH"
STARTLOOP = 1
'
'             RECORD INITIAL ENCODER VALUE
```

```
'
'               COPY ENCODER COUNTERS TO HOLDING REGISTERS
'
    OUT C,1
'
'               SELECT COUNTER X REGISTER
'
    OUT P,1
'
'               BEGIN LOOP FOR INPUT
'
FOR I = 1 TO 6
    X(I) = INP(C)
NEXT I
'
'               CONVERT TO DECIMAL VALUE
'
XV=X(6)+X(5)*16+X(4)*256+X(3)*4096+X(2)*65536!+X(1)*1048576!
'
'               SET FOR A PLUS/MINUS READING
'
IF XV>8388607! THEN XV=XV-1.677722E+07
POSIDAT(1,2) = XV
'
'               COMBINE AIR AND HYD VALVE ORDERS
'
ORDER% = AV% OR HV%
'
'               HARDWARE INTERFACE USES NEGATIVE TRUE LOGIC...
'               NEGATE MOVEMENT ORDER AND SEND OUT TO SOLENOIDS

'
NORDER% = NOT(ORDER%) AND &HFF
POSIDAT(1,1) = 0.0
MTIMER
OUT SOLENOIDS,NORDER%

'
'               RECORD THE ENCODER AND TIME VALUES DURING THE
MOVE
'
COUNT% = 2
DO WHILE STARTLOOP = 1
    POSIDAT(COUNT%,1) = MTIMER/1E6
    OUT C,1
    OUT P,1
    MTIMER
    FOR I = 1 TO 6
        X(I) = INP(C)
    NEXT I
XV=X(6)+X(5)*16+X(4)*256+X(3)*4096+X(2)*65536!+X(1)*1048576!
    IF XV>8388607! THEN XV=XV-1.677722E+07
    POSIDAT(COUNT%,2) = XV
```

```
    INCR COUNT%
    A = INP(&H331) AND 16
    IF A = 0 THEN GOTO 300
LOOP
300 OUT SOLENOIDS,255
TICK% = COUNT% - 1
PRINT
PRINT "THE NUMBER OF ENCODER READS WAS";TICK%
POSIT = POSIDAT(TICK%,2)
PRINT "NEW POSITION IS";POSIT
PRINT "SEND DATA TO A FILE(Y OR N)?"
INPUT Q$
IF Q$ = "N" THEN GOTO 100
PRINT "ENTER DOS FILENAME FOR DATA OUTPUT"
INPUT DATAFILE$
OPEN DATAFILE$ FOR APPEND AS #1
PRINT #1, TICK%
FOR I = 1 TO TICK%
   PRINT #1,USING "########.#####";POSIDAT(I,1),POSIDAT(I,2)
NEXT I
CLOSE #1
GOTO 100
400 END
```

```
'                       MANIP.BAS
'
' THIS PROGRAM WAS CREATED TO MANIPULATE THE TEST DATA
' GENERATED BY THE TESTOP.BAS PROGRAM FOR DYNAMIC
' MODEL VERIFICATION.
'                               JDI   11/12/88.
'
100 PRINT "RUN DATA MANIPULATION PROGRAM(1 = Y, 2 = N)?
INPUT Q1
IF Q1 <> 1 THEN GOTO 300
CLS
PRINT "THIS PROGRAM MANIPULATES ENCODER DATA ACQUIRED"
PRINT "USING TESTOP.BAS"
PRINT
PRINT "ENTER THE DOS FILENAME OF THE DATA FILE"
INPUT DATAFILE$
OPEN DATAFILE$ FOR INPUT AS #1
INPUT #1,COUNT%
'
'             DIMENSION ARRAYS TO MANIPULATE ENCODER DATA
'
OPTION BASE 1
DIM POSIDAT(COUNT%,9)
'
'             READ DELTA T AND ENCODER POSITS FROM DOS FILE
'
FOR I = 1 TO COUNT%
    INPUT #1,POSIDAT(I,1)
    INPUT #1,POSIDAT(I,2)
NEXT I
CLOSE #1
'
'             DEFINE PI FOR CONVERSION OF COUNTS TO RADIANS
'
PI = 4.0 * ATN(1.0)
'
'             ADD UP THE TIME INCREMENTS FOR ELAPSED TIME,
'             COMPUTE POSITIONS AND INSTANTANEOUS VELOCITIES
'             IN TERMS OF DEGREES OF MOTION THETA AND
'              DISPLACEMENT OF THE HYDRAULIC PISTON X1, AND
'             SEND THE STORED DATA TO A SEQUENTIAL FILE
'
POSIDAT(1,3) = 0.0
POSIDAT(1,4) = POSIDAT(1,2) * ((2.0*PI)/(8192.0))
POSIDAT(1,5) = POSIDAT(1,4) - 0.4693981
POSIDAT(1,6) = POSIDAT(1,5) * 180.0/PI
POSIDAT(1,7) = 0.0
C1 = 119.24275 - 55.6182*COS(1.320697-POSIDAT(1,5))
POSIDAT(1,8) = 11.460 - (C1^(0.5))
POSIDAT(1,9) = 0.0
FOR I = 2 TO COUNT%
    J = I - 1
    POSIDAT(I,3) = POSIDAT(I,1) + POSIDAT(J,3)
```

```
      POSIDAT(I,4) = POSIDAT(I,2) * ((2.0*PI)/(8192.0))
      POSIDAT(I,5) = POSIDAT(I,4) - 0.4693981
      POSIDAT(I,6) = POSIDAT(I,5) * 180.0/PI
      DELTATHETA = POSIDAT(I,6) - POSIDAT(J,6)
      DELTAT = POSIDAT(I,1)
      IF DELTAT = 0.0 THEN
          POSIDAT(I,7) = 0.0
      ELSE
          POSIDAT(I,7) = ABS(DELTATHETA/DELTAT)
      END IF
      C1 = 119.24275 - 55.6182*COS(1.320697-POSIDAT(I,5))
      POSIDAT(I,8) = 11.460 - (C1^(0.5))
      DELTAX1 = POSIDAT(I,8) - POSIDAT(J,8)
      IF DELTAT = 0.0 THEN
          POSIDAT(I,9) = 0.0
      ELSE
          POSIDAT(I,9) = ABS(DELTAX1/DELTAT)
      END IF
NEXT I
PRINT
PRINT "SEND DATA TO A FILE(1 = Y, 0 = N)?"
INPUT Q2
IF Q2 <> 1 THEN GOTO 200
PRINT "ENTER DOS FILENAME FOR DATA OUTPUT"
INPUT MANFILE$
OPEN MANFILE$ FOR APPEND AS #2
PRINT #2,"        TIME";"         THETA";"         THETADOT";
PRINT #2,"         X1";"          X1DOT"
PRINT #2,
FOR I = 1 TO COUNT%
   PRINT #2,USING "#######.#####";POSIDAT(I,3),POSIDAT(I,6);
   PRINT #2,USING "#######.#####";POSIDAT(I,7),POSIDAT(I,8);
   PRINT #2,USING "#######.#####";POSIDAT(I,9)
NEXT I
CLOSE #2
ERASE POSIDAT
GOTO 100
200 PRINT "        TIME";"         THETA";"         THETADOT";
PRINT "        X1";"          X1DOT"
PRINT
FOR I = 1 TO COUNT%
   PRINT USING "#######.#####";POSIDAT(I,3),POSIDAT(I,6);
   PRINT USING "#######.#####";POSIDAT(I,7),POSIDAT(I,8);
   PRINT USING "#######.#####";POSIDAT(I,9)
NEXT I
ERASE POSIDAT
GOTO 100
300 END
```

```
**********************************************************************
*                         MANIP FORTRAN A1                           *
*                                                                    *
* THIS PROGRAM WAS CREATED TO MANIPULATE THE TEST DATA               *
* GENERATED BY THE TESTOP.BAS PROGRAM FOR DYNAMIC MODEL              *
* VERIFICATION.  FORTRAN VERSION FOR MAINFRAME USE TO                *
* MANIPULATE FILES LARGER THAN PC CAN HANDLE.                        *
*                                            JDI   11/12/88.         *
**********************************************************************
*
      REAL PI,C1,DTH,DT,DX1,A,B,C,D,E
      INTEGER I,J,Q1,Q2,COUNT
      DIMENSION POSDAT(5000,9)
100   PRINT *,'YOU MUST HAVE THE DATA TO BE WORKED ON IN A
  &FILE'
      PRINT *,'CALLED "FILE NOVEL A1" ON YOUR DISK.'
      PRINT *,'RUN DATA MANIPULATION PROGRAM(1 = Y, 2 = N)?'
      READ *, Q1
      IF (Q1 .NE. 1) GO TO 800
      PRINT *
      PRINT *,'THIS PROGRAM MANIPULATES ENCODER DATA
     &ACQUIRED'
      PRINT *,'USING TESTOP.BAS'
      PRINT *
      OPEN (UNIT = 10, FILE = 'NOVEL', STATUS = 'OLD')
      READ (10,*) COUNT
*
*            READ DELTA T AND ENCODER POSITS FROM RAW DATA
*            FILE
*
      DO 200 I = 1,COUNT
         READ(10,*) POSDAT(I,1),POSDAT(I,2)
200   CONTINUE
      CLOSE (10)
*
*            DEFINE PI FOR CONVERSION OF COUNTS TO RADIANS
*
      PI = 4.0 * ATAN(1.0)
*
*            ADD UP THE TIME INCREMENTS FOR ELAPSED TIME,
*            COMPUTE POSITIONS AND INSTANTANEOUS VELOCITIES
*            IN TERMS OF DEGREES OF MOTION THETA AND
*            DISPLACEMENT OF THE HYDRAULIC PISTON X1, AND
*            SEND THE STORED DATA TO A SEQUENTIAL FILE
*
      POSDAT(1,3) = 0.0
      POSDAT(1,4) = POSDAT(1,2) * ((2.0*PI)/(8192.0))
      POSDAT(1,5) = POSDAT(1,4) - 0.4693981
      POSDAT(1,6) = POSDAT(1,5) * 180.0/PI
      POSDAT(1,7) = 0.0
      C1 = 119.24275 - 55.6182*COS(1.320697-POSDAT(1,5))
      POSDAT(1,8) = 11.460 - (C1**(0.5))
      POSDAT(1,9) = 0.0
```

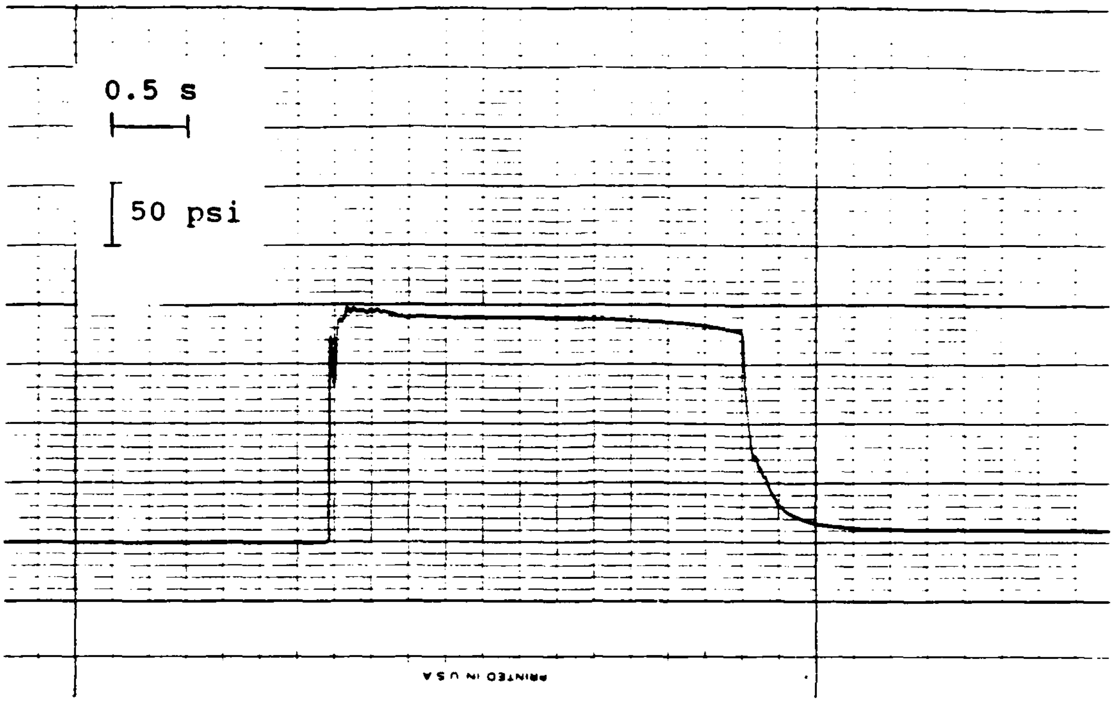
```
      DO 300 I = 2,COUNT
         J = I - 1
         POSDAT(I,3) = POSDAT(I,1) + POSDAT(J,3)
         POSDAT(I,4) = POSDAT(I,2) * ((2.0*PI)/(8192.0))
         POSDAT(I,5) = POSDAT(I,4) - 0.4693981
         POSDAT(I,6) = POSDAT(I,5) * 180.0/PI
         DTH = POSDAT(I,6) - POSDAT(J,6)
         DT = POSDAT(I,1)
         IF (DT .EQ. 0.0) THEN
            POSDAT(I,7) = 0.0
         ELSE
            POSDAT(I,7) = ABS(DTH/DT)
         END IF
         C1 = 119.24275 - 55.6182*COS(1.320697-POSDAT(I,5))
         POSDAT(I,8) = 11.460 - (C1**(0.5))
         DX1 = POSDAT(I,8) - POSDAT(J,8)
         IF (DT .EQ. 0.0) THEN
            POSDAT(I,9) = 0.0
         ELSE
            POSDAT(I,9) = ABS(DX1/DT)
         END IF
300      CONTINUE
      PRINT *
      PRINT *,'SEND DATA TO A FILE(1 = Y, 0 = N)?'
      READ *,Q2
      IF (Q2 .NE. 1) GO TO 600
      PRINT *,'OUTPUT IS IN A FILE NAMED "FILE PRCSSD A1" '
      OPEN (UNIT = 20, FILE = 'PRCSSD', STATUS = 'NEW')
      WRITE(20,*)'         TIME','         THETA','      THETADOT',
     &'        X1','           X1DOT'
      WRITE(20,*)
400   FORMAT(1X,5F12.5)
      DO 500 I = 1,COUNT
         A = POSDAT(I,3)
         B = POSDAT(I,6)
         C = POSDAT(I,7)
         D = POSDAT(I,8)
         E = POSDAT(I,9)
         WRITE(20,400) A,B,C,D,E
500   CONTINUE
      CLOSE (20)
      GOTO 800
600   PRINT *,'        TIME','         THETA','          THETADOT',
     &'        X1','           X1DOT'
      PRINT *
      DO 700 I = 1,COUNT
         A = POSDAT(I,3)
         B = POSDAT(I,6)
         C = POSDAT(I,7)
         D = POSDAT(I,8)
         E = POSDAT(I,9)
         WRITE(*,400) A,B,C,D,E
```

```
700   CONTINUE
800   END
```

APPENDIX E

HYDRAULIC PRESSURE STRIP CHART RECORDINGS

Recordings of the hydraulic pressure difference for raising and lowering motions and the load conditions of no load, two pounds, and five pounds are shown in Figures E-1 through E-12. The worst case pressure reading for orifice one, five pound load, lowering motion, is shown as Figure E-13.



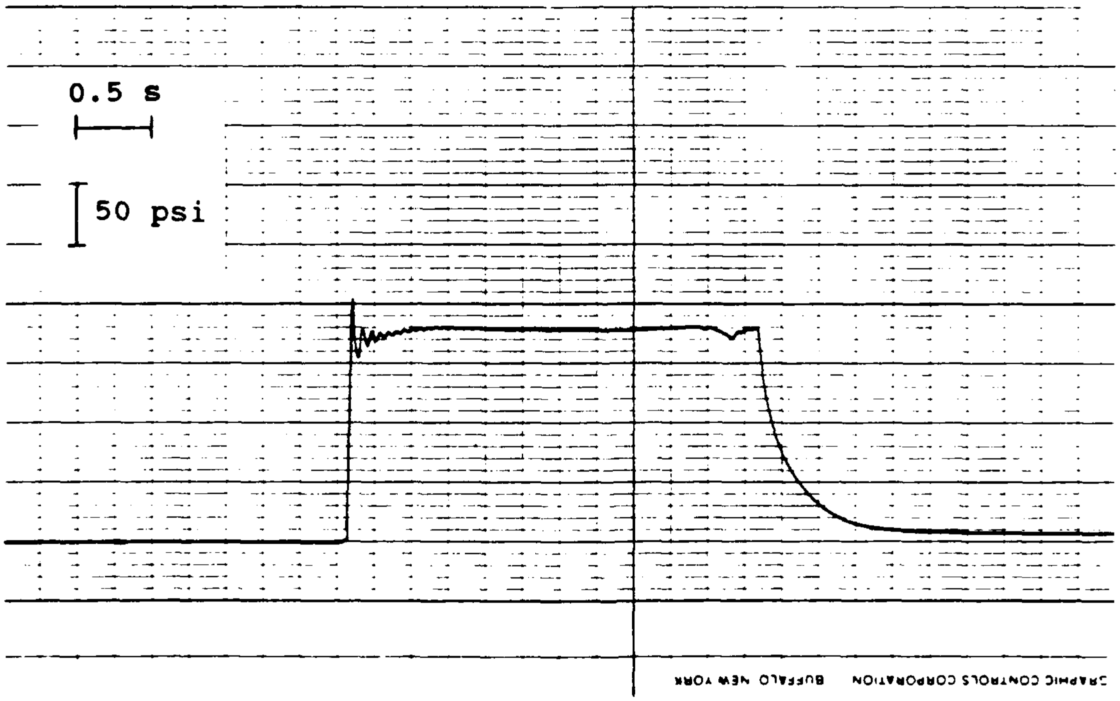


Figure E-1 Pressure Difference, No Load, Orifice 1.
Top Lowering, Bottom Raising

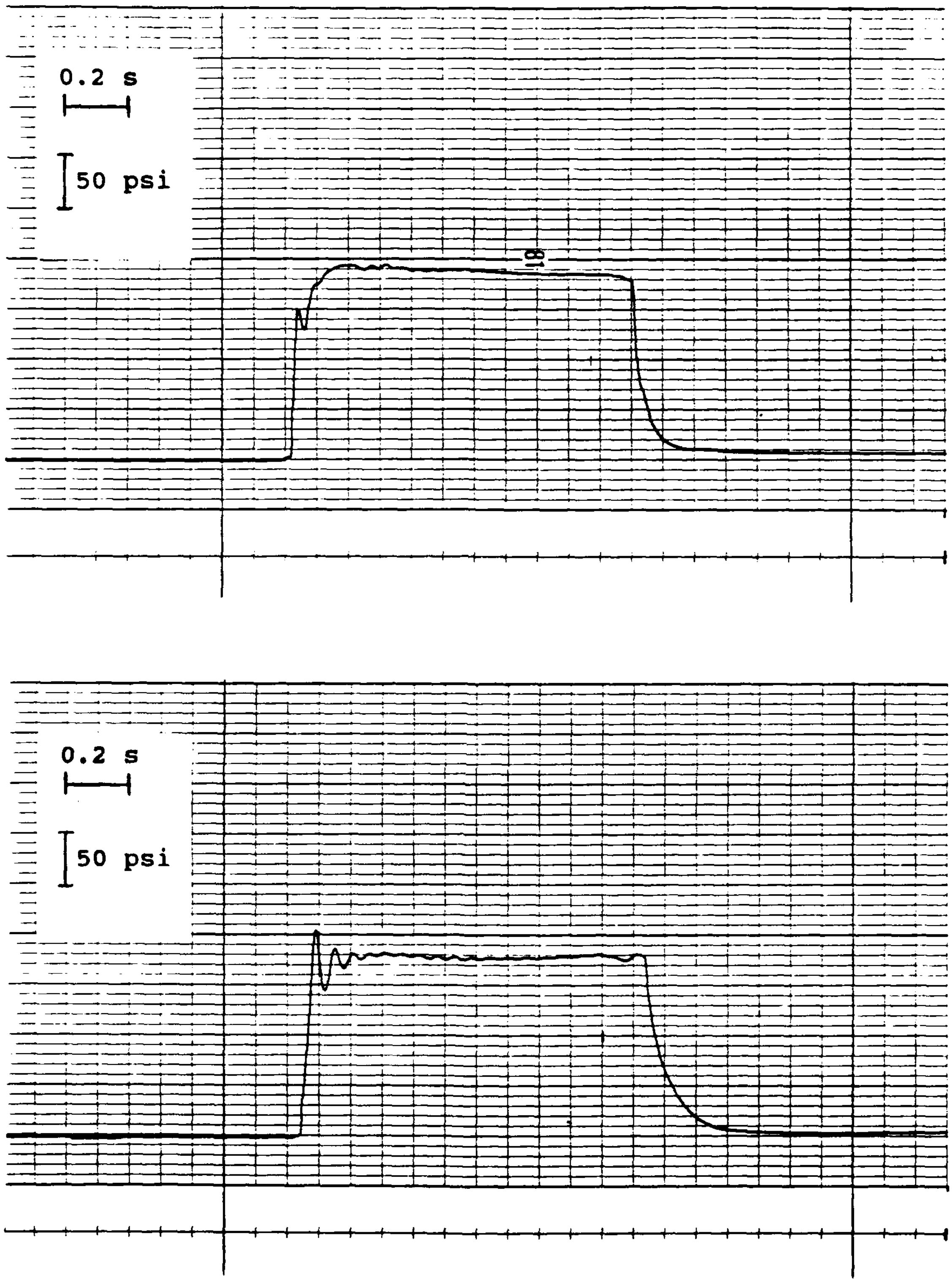


Figure E-2 Pressure Difference, No Load, Orifice 2. Top Lowering, Bottom Raising

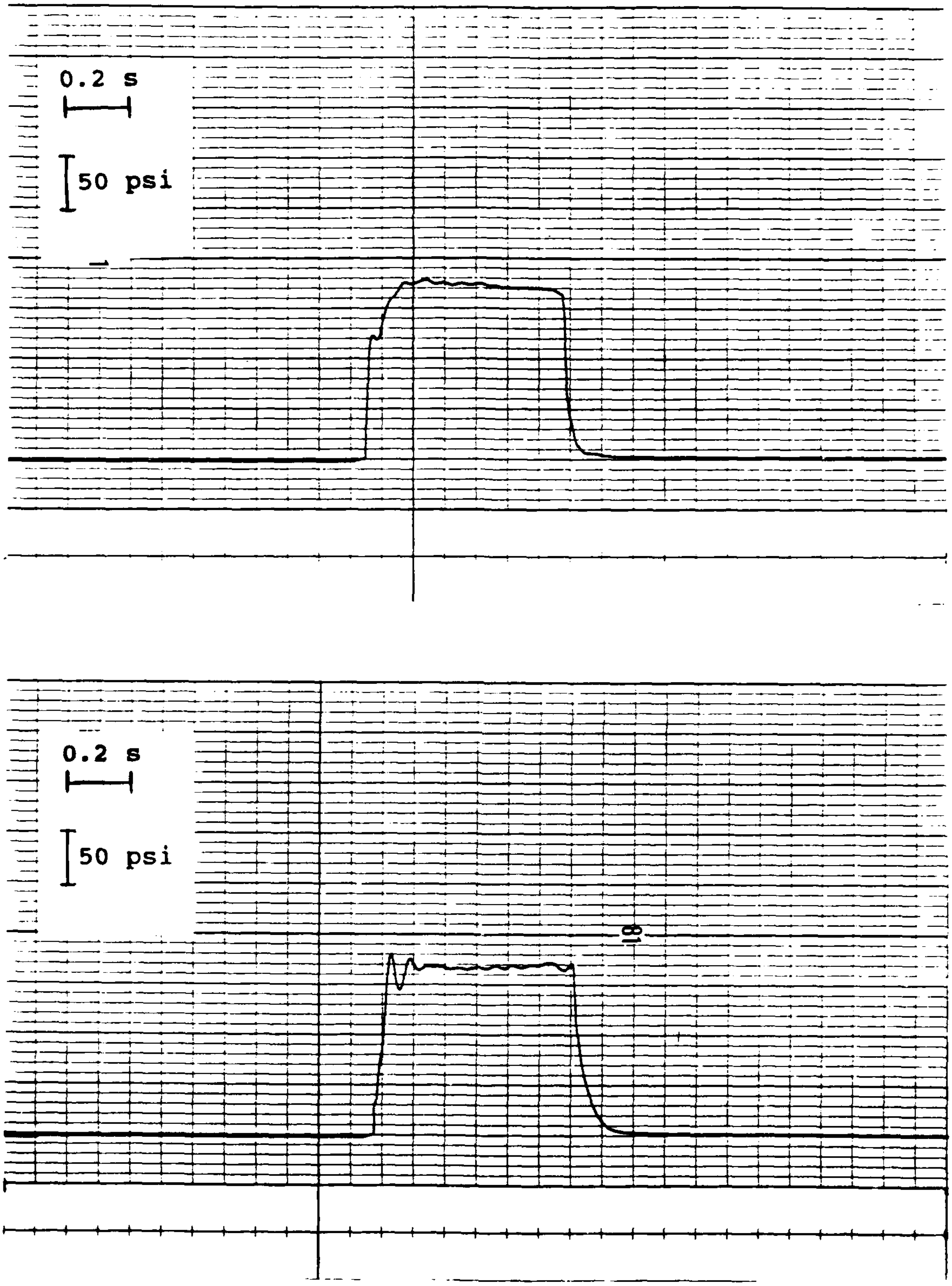
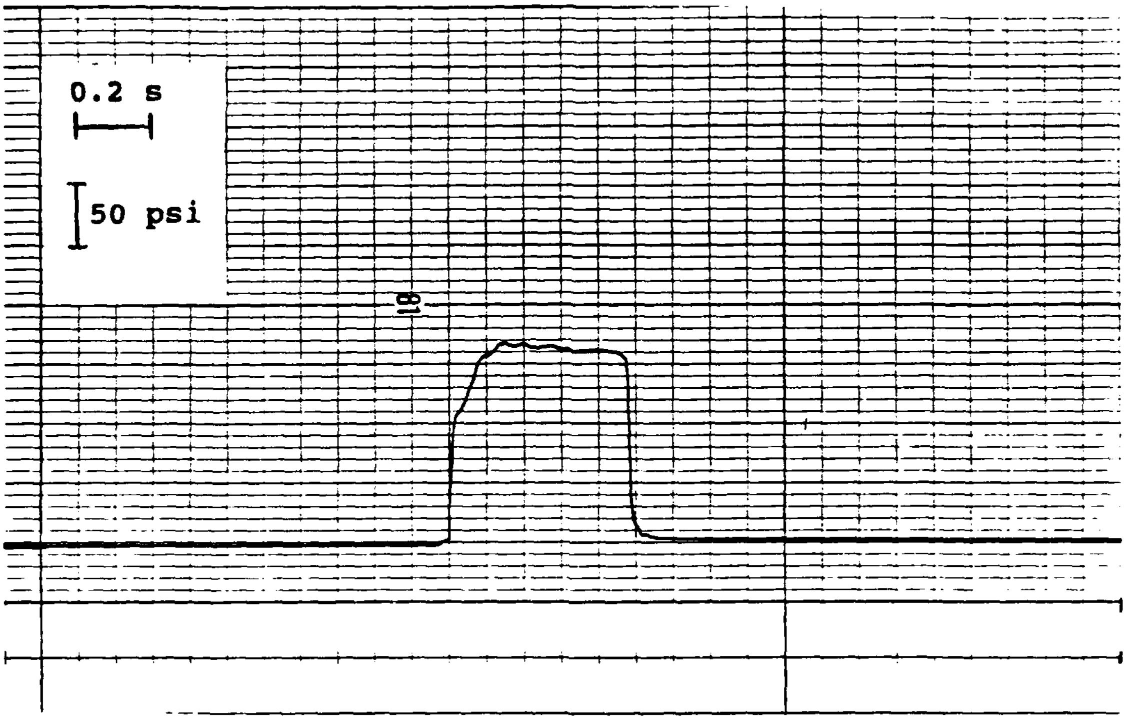


Figure E-3 Pressure Difference, No Load, Orifice 3.
Top Lowering, Bottom Raising



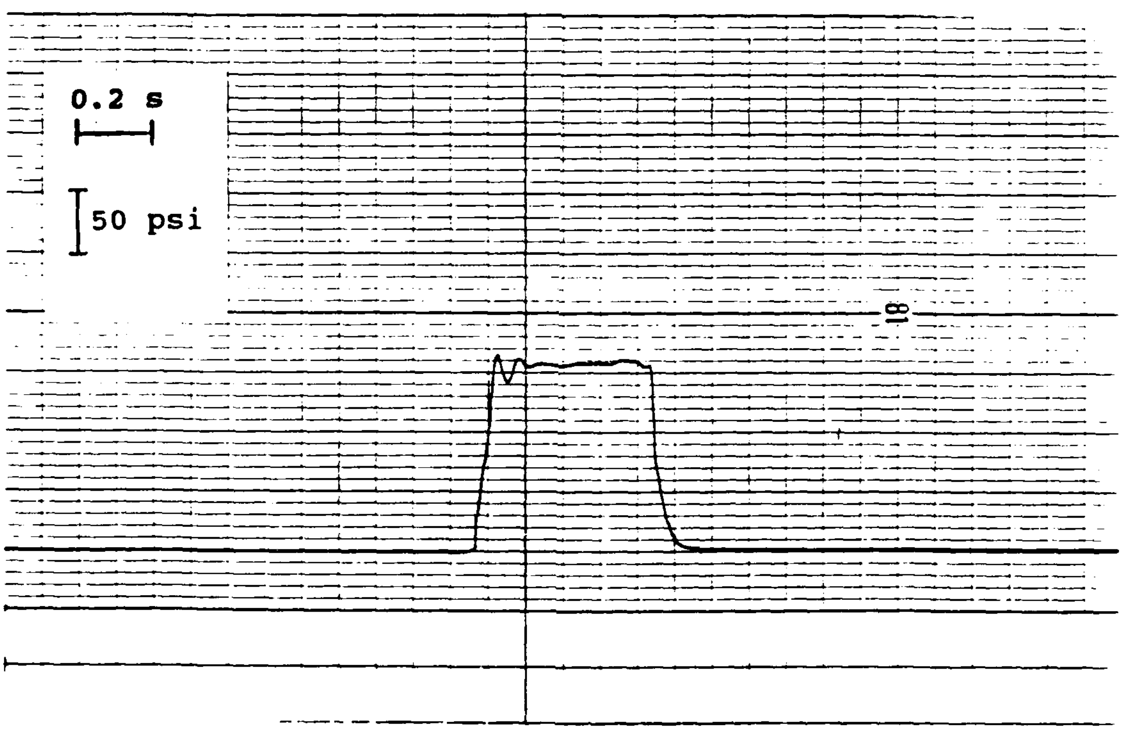
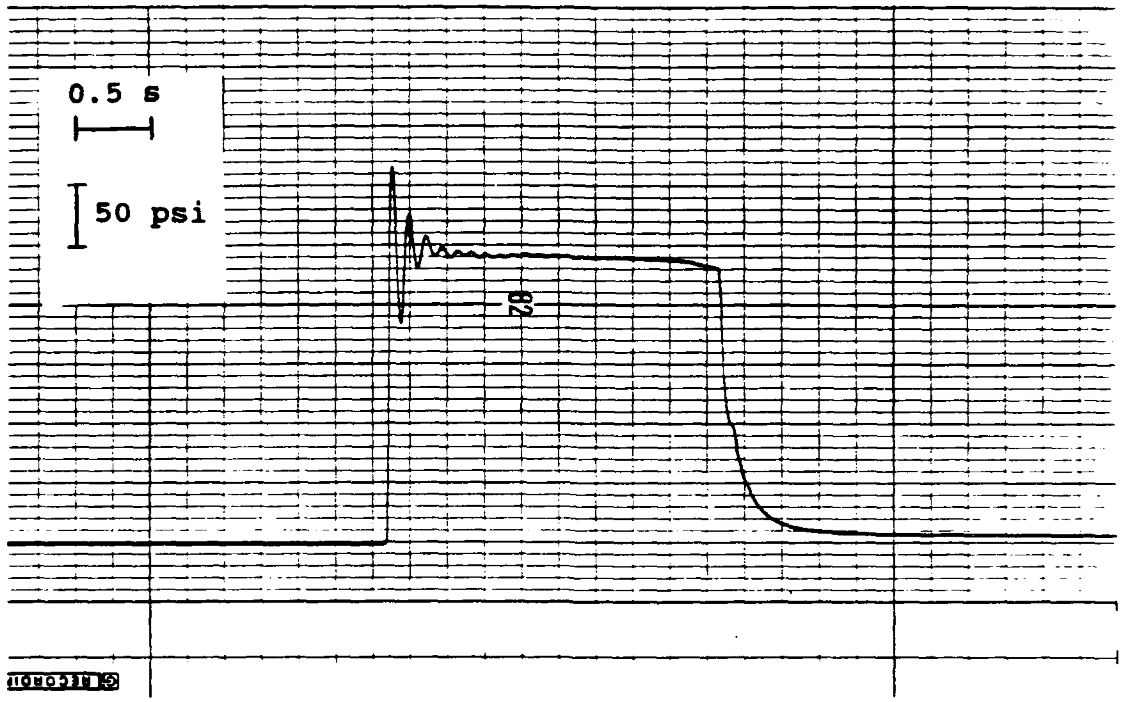


Figure E-4 Pressure Difference, No Load, Orifice 4.
Top Lowering, Bottom Raising



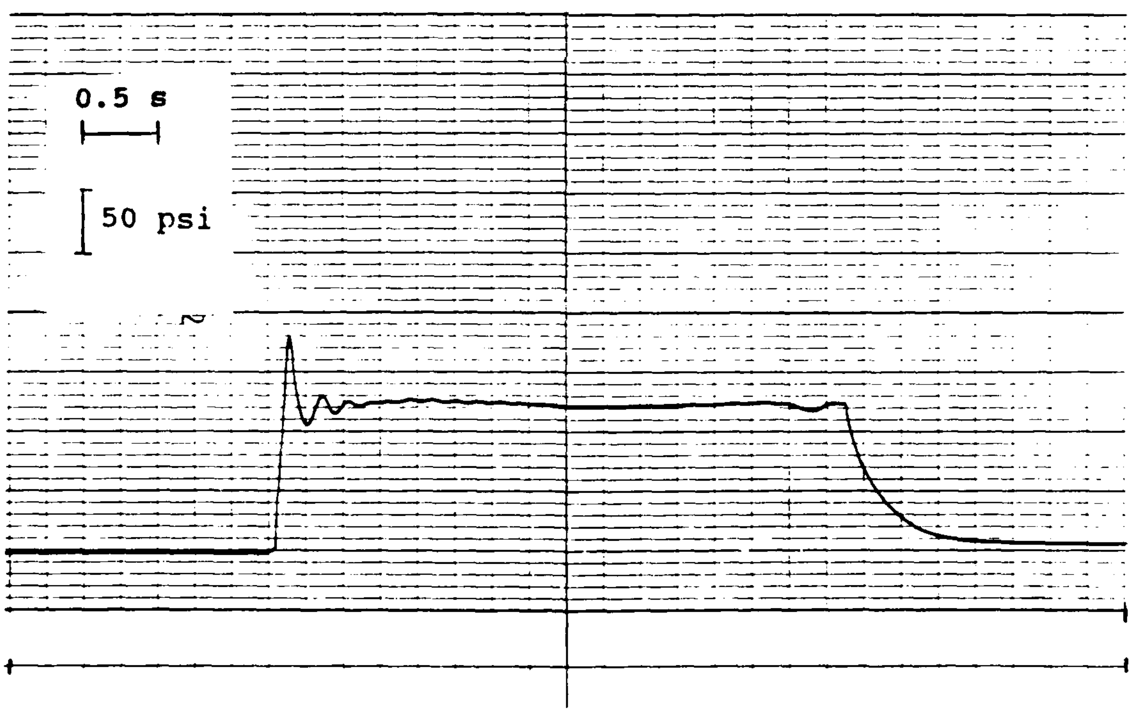
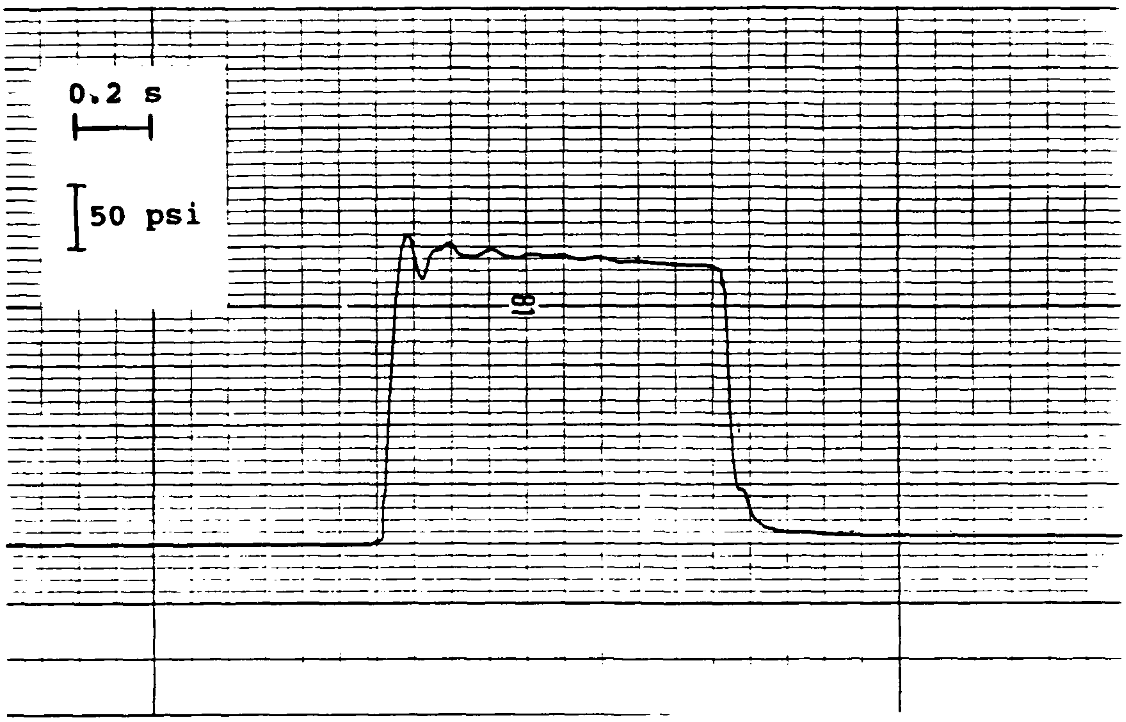


Figure E-5 Pressure Difference, 2 Lb Load, Orifice 1.
Top Lowering, Bottom Raising



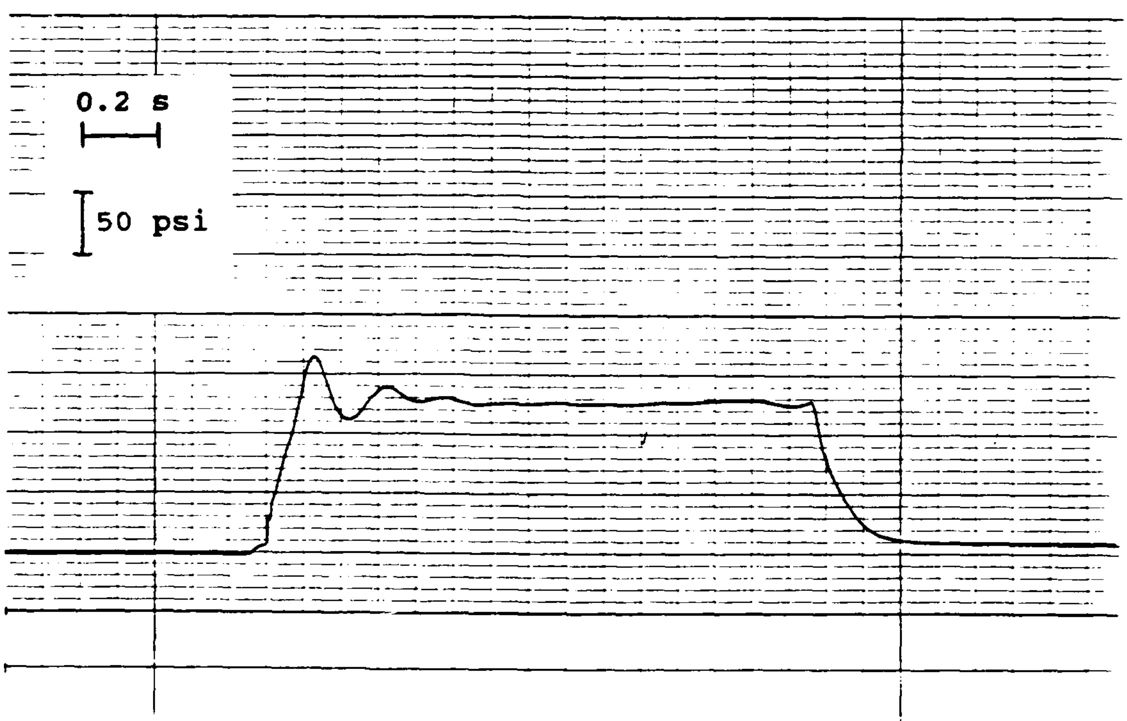
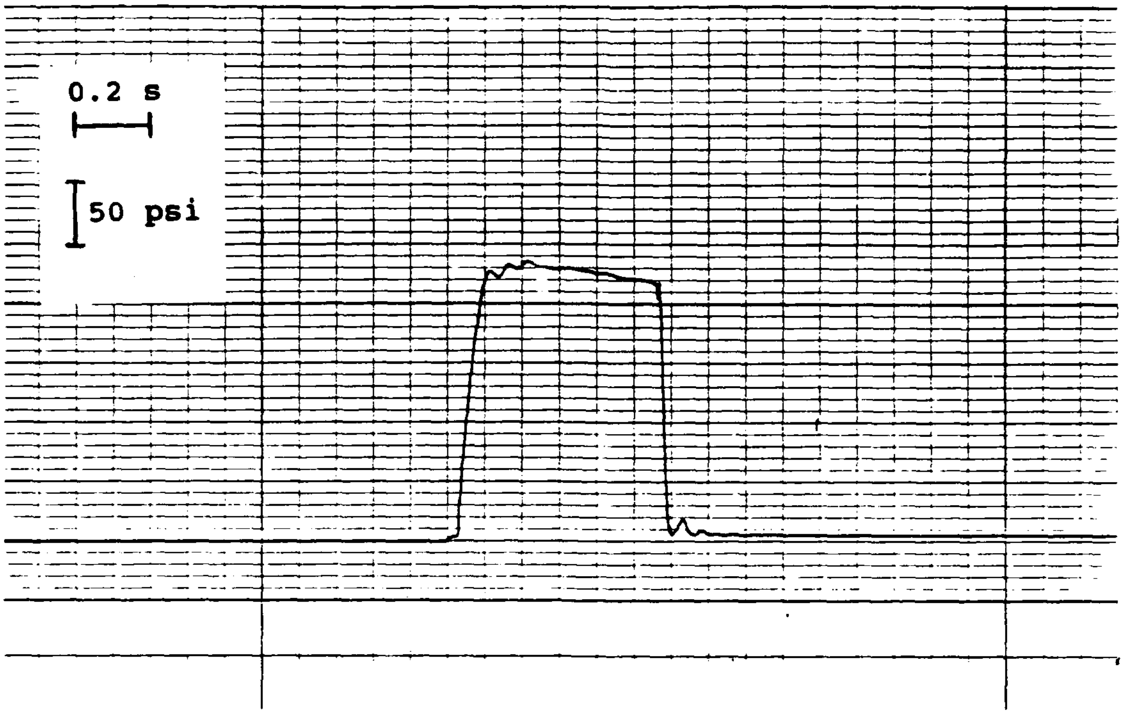


Figure E-6 Pressure Difference, 2 Lb Load, Orifice 2.
Top Lowering, Bottom Raising



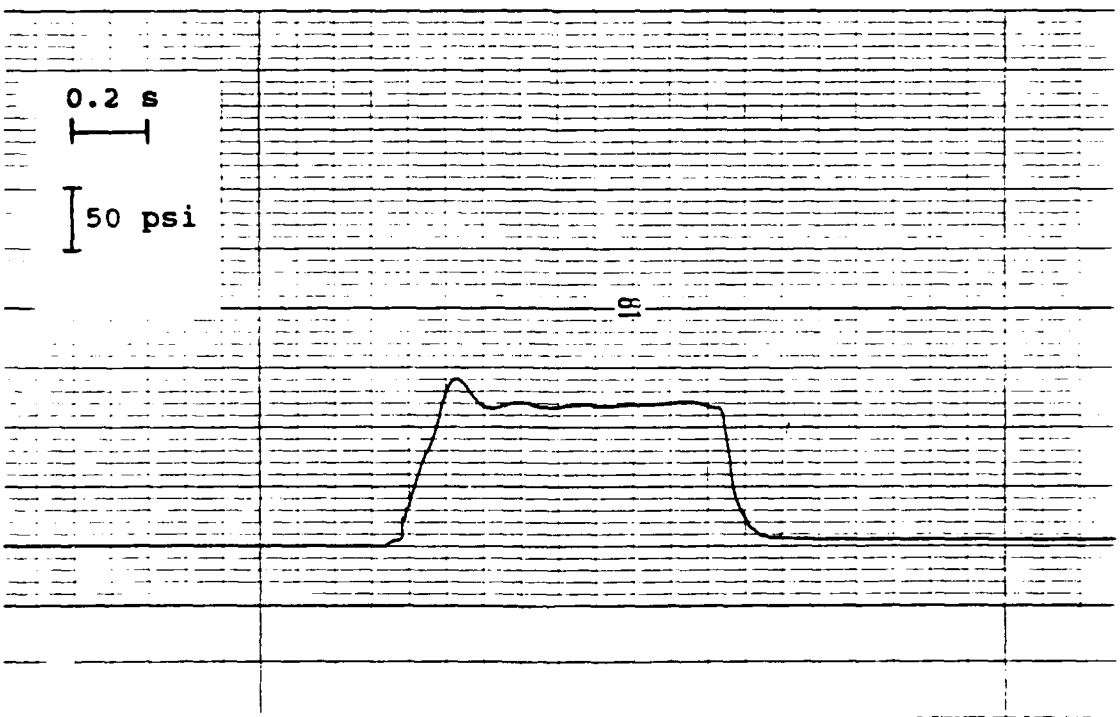


Figure E-7 Pressure Difference, 2 Lb Load, Orifice 3. Top Lowering, Bottom Raising

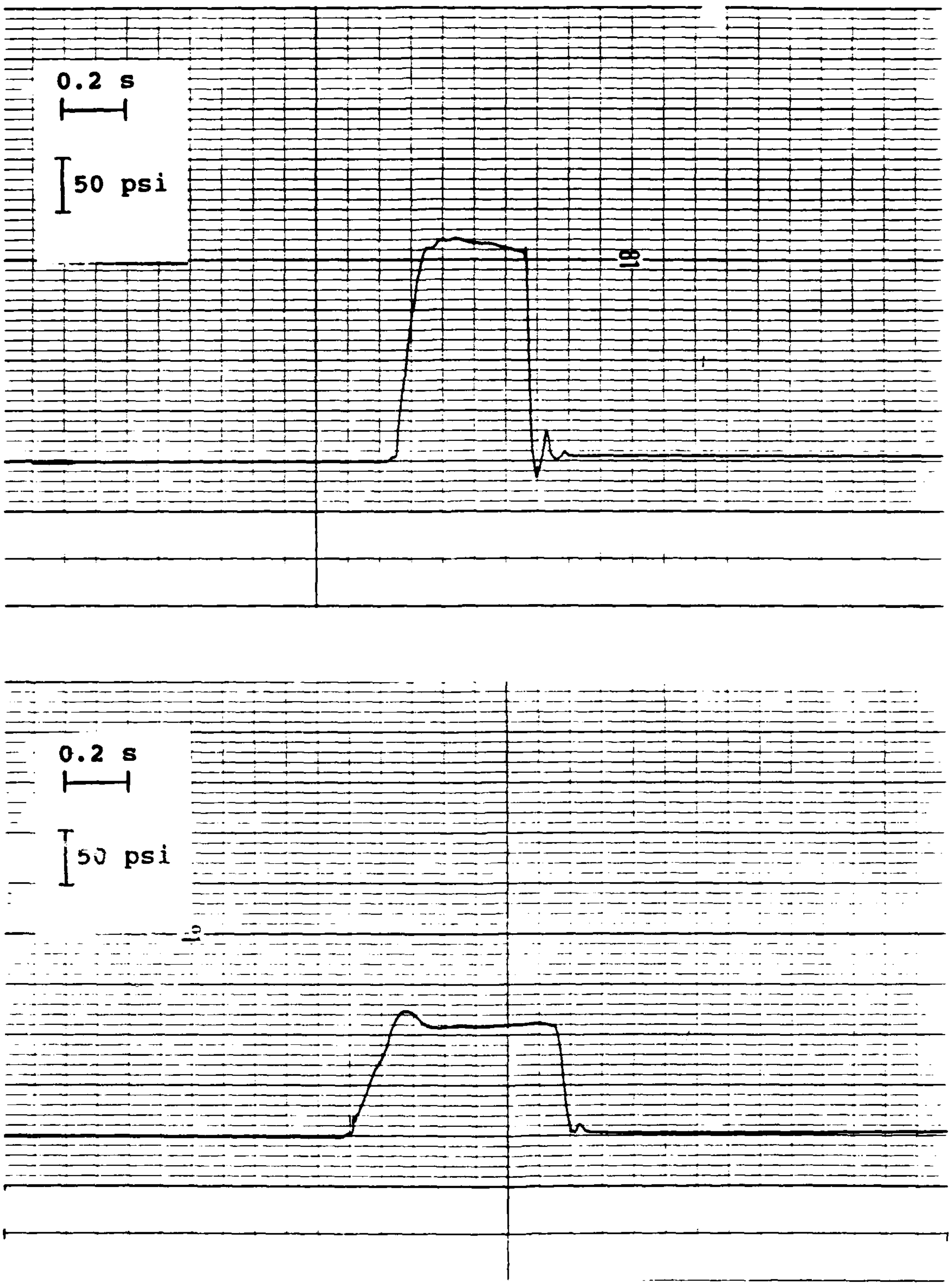
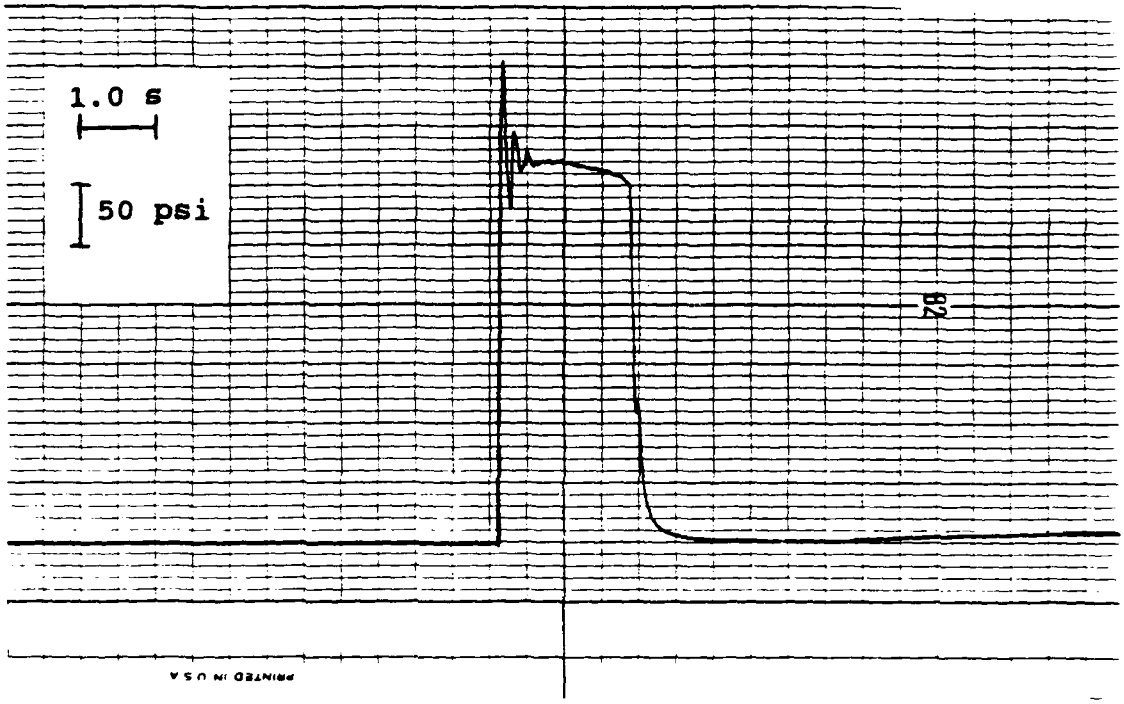


Figure E-8 Pressure Difference, 2 Lb Load, Orifice 4.
Top Lowering, Bottom Raising



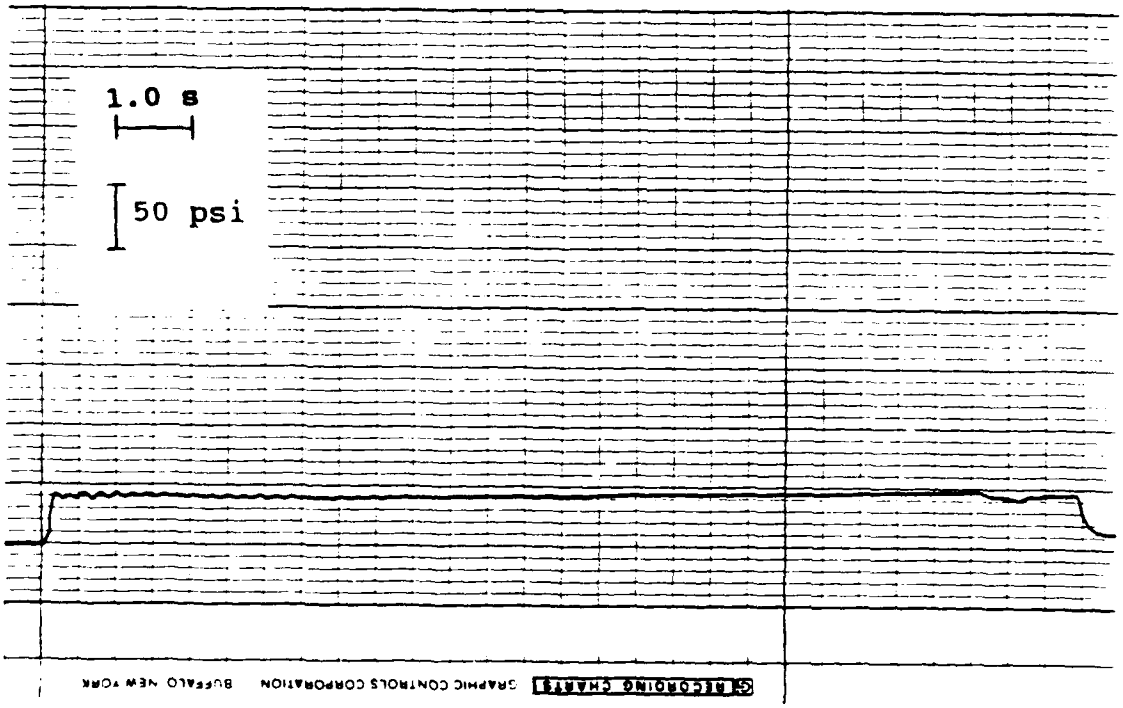


Figure E-9 Pressure Difference, 5 Lb Load, Orifice 1.
Top Lowering, Bottom Raising

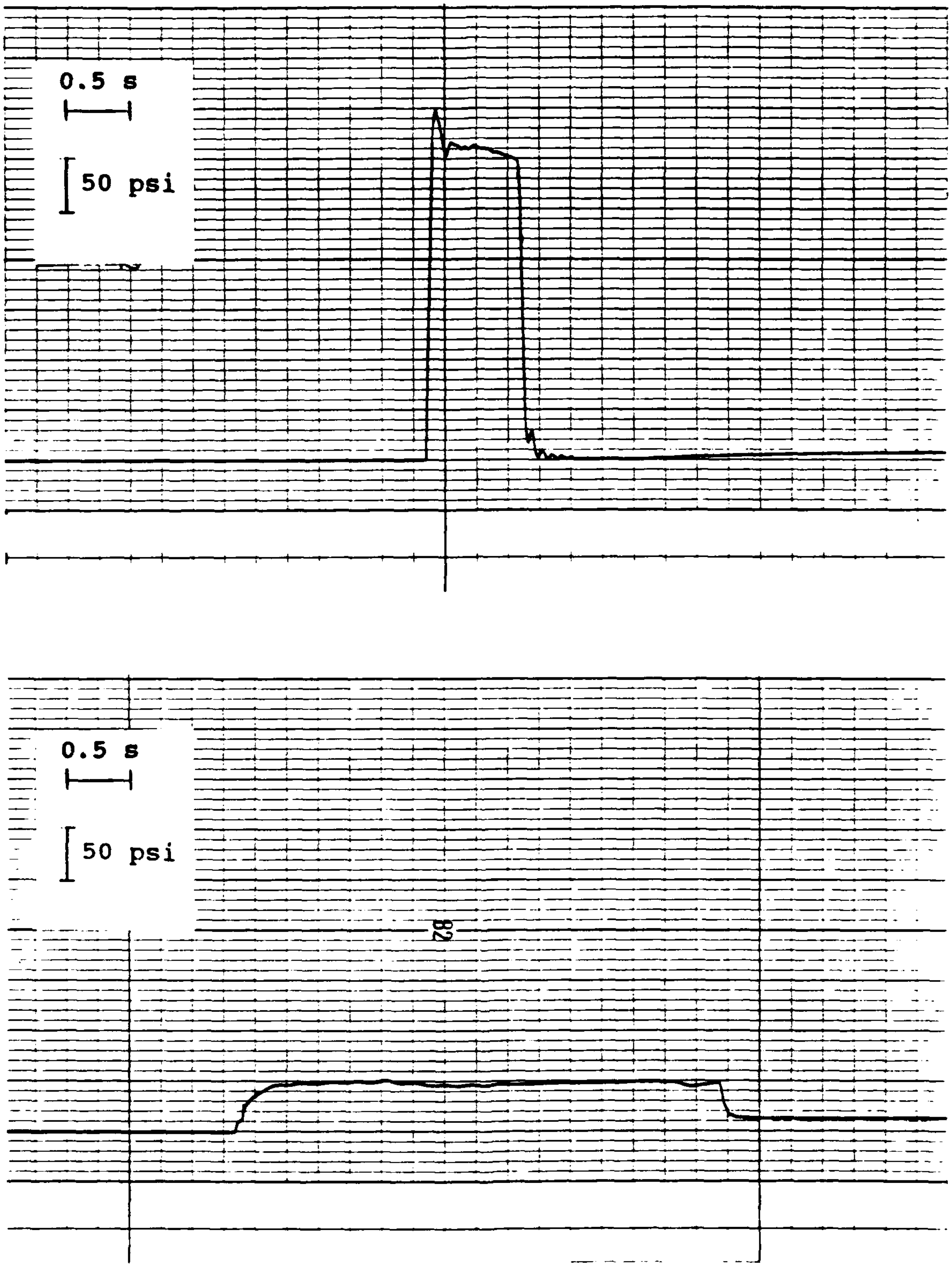


Figure E-10 Pressure Difference, 5 Lb Load, Orifice 2.
Top Lowering, Bottom Raising

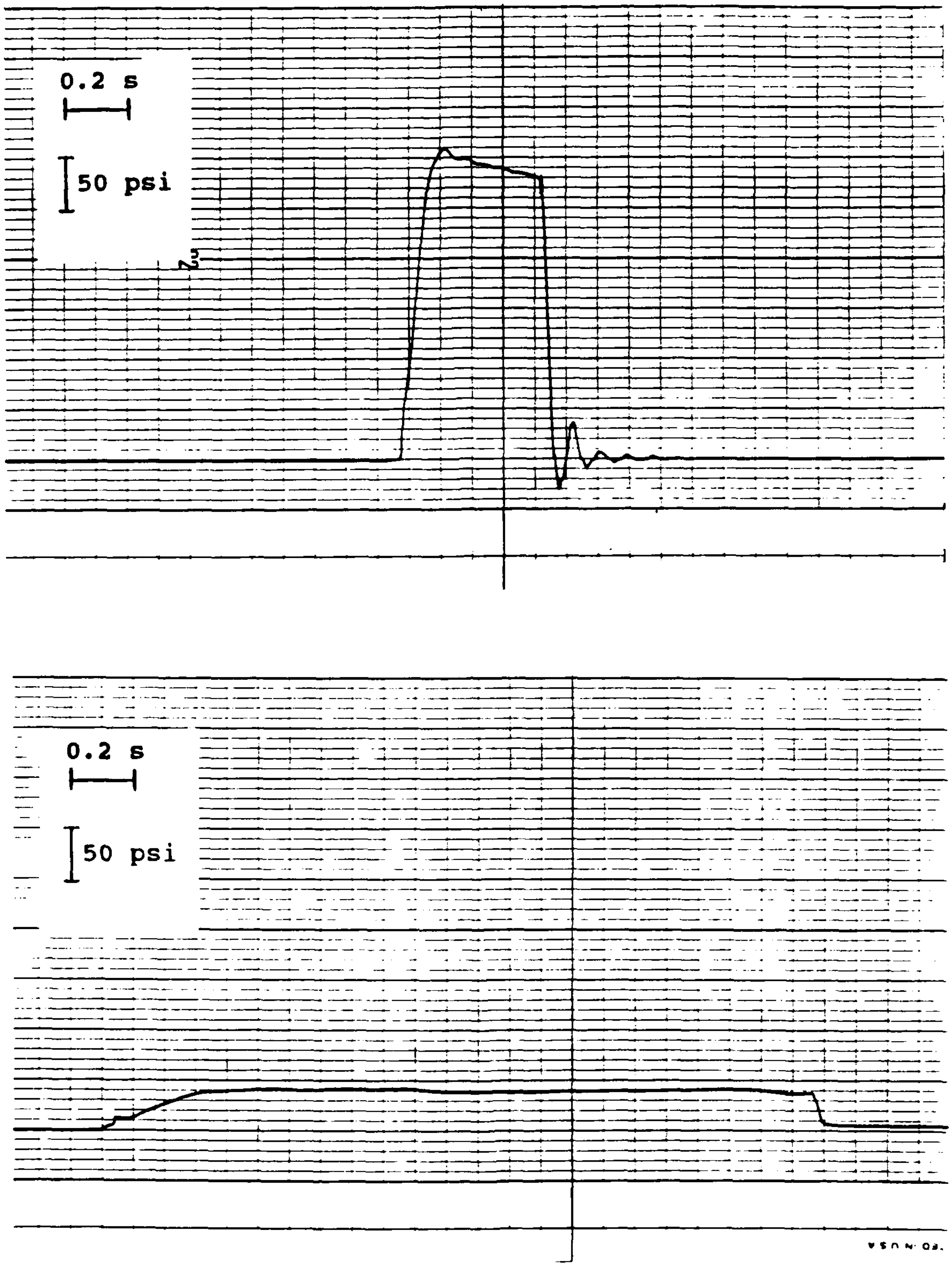


Figure E-11 Pressure Difference, 5 Lb Load, Orifice 3.
Top Lowering, Bottom Raising

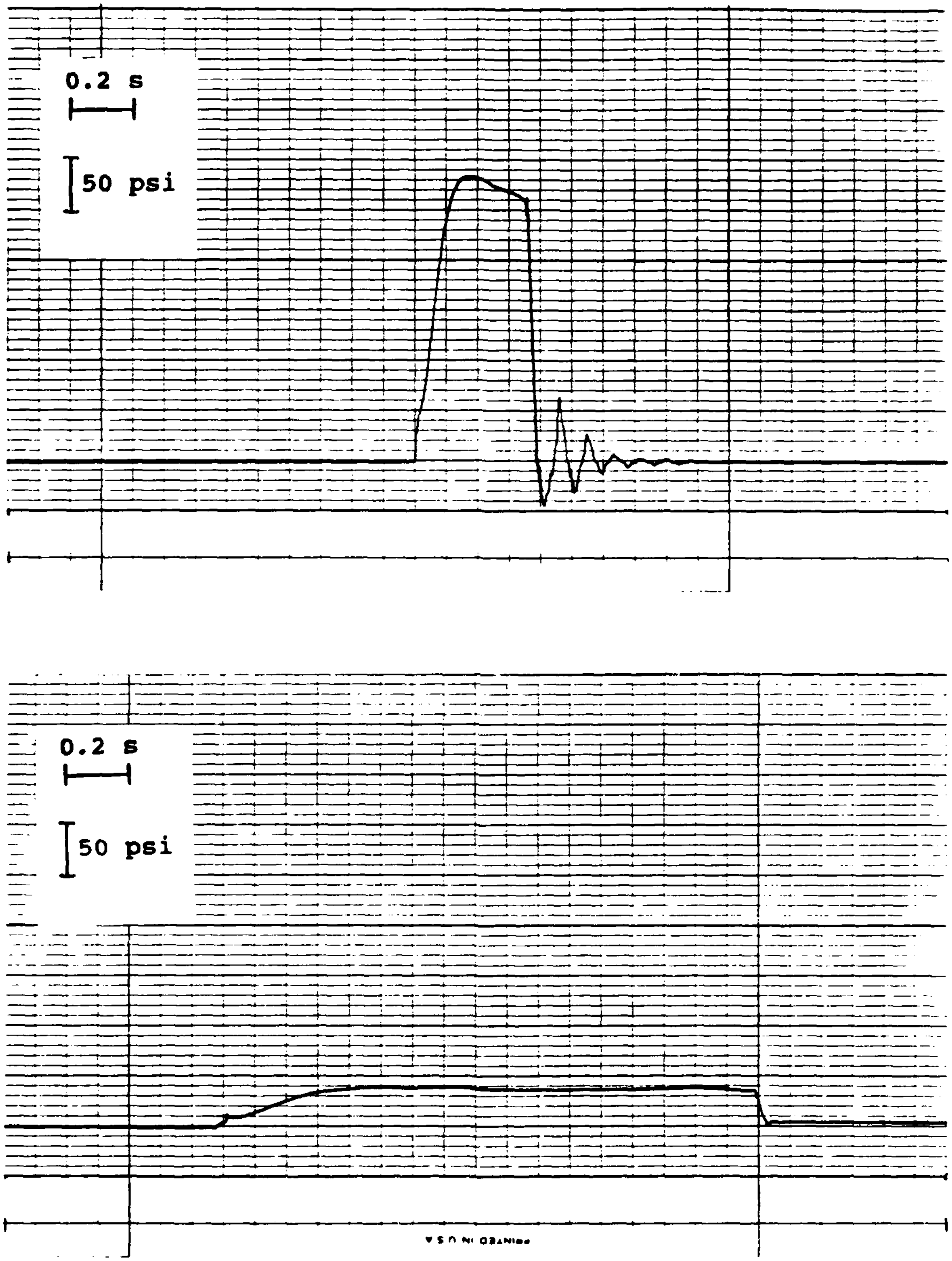


Figure E-12 Pressure Difference, 5 Lb Load, Orifice 4.
Top Lowering, Bottom Raising

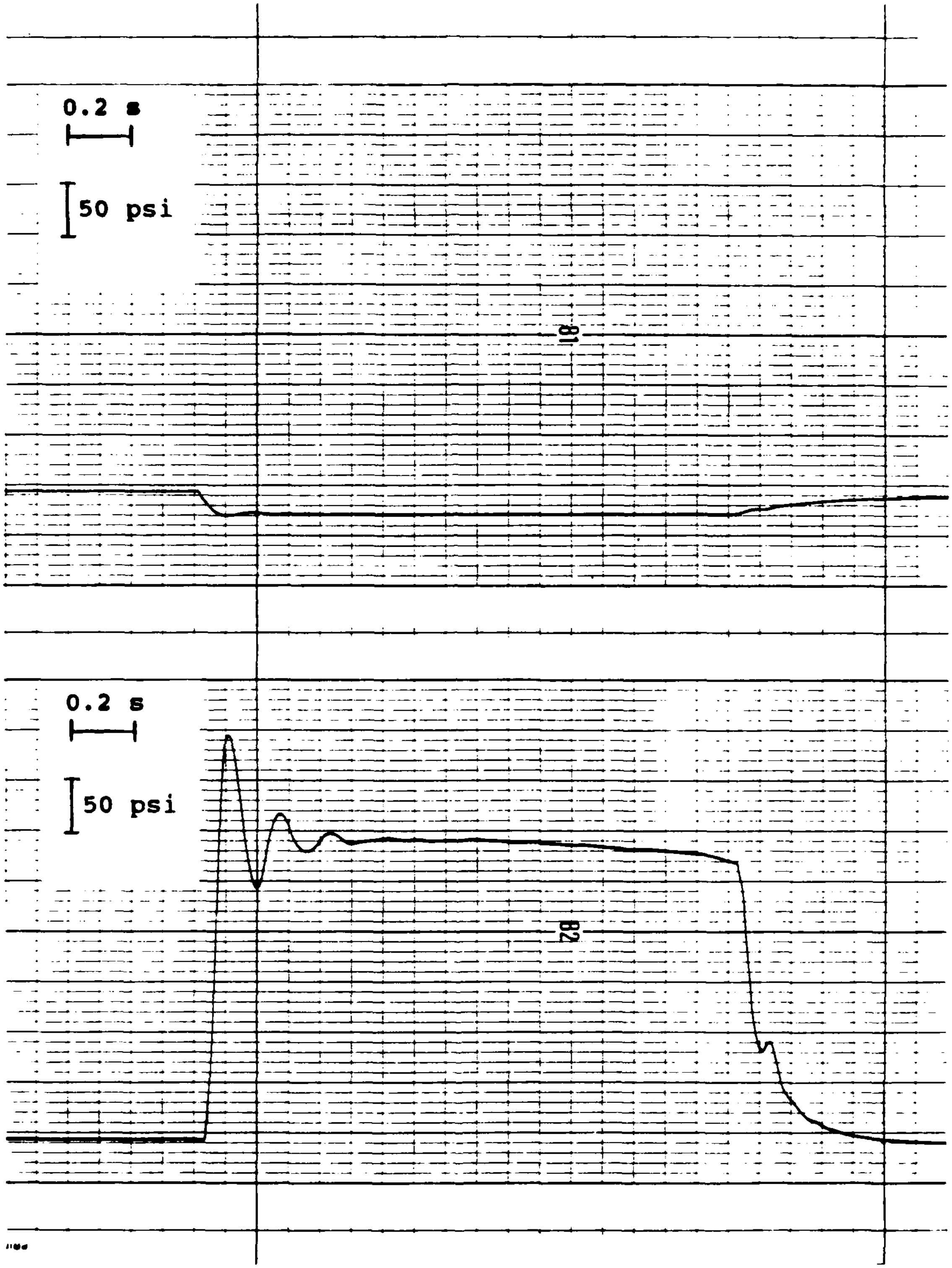


Figure E-13 Cylinder Pressures, 5 Lb Load, Orifice 1, Lowering. Top P_4, Bottom P_5

APPENDIX F

HYDRAULIC PISTON POSITION VERSUS TIME

Figures F-1 through F-12 represent hydraulic piston position X_1 versus time for each orifice under the specified load conditions. The zero position for coordinate X_1 was taken to be when the Novel Actuator Test Bed mechanism was in the fully lowered position, and the hydraulic cylinder rod was at its maximum extension. Accordingly, raising motions are indicated by curves with negative slope, and lowering motions are curves with positive slopes.

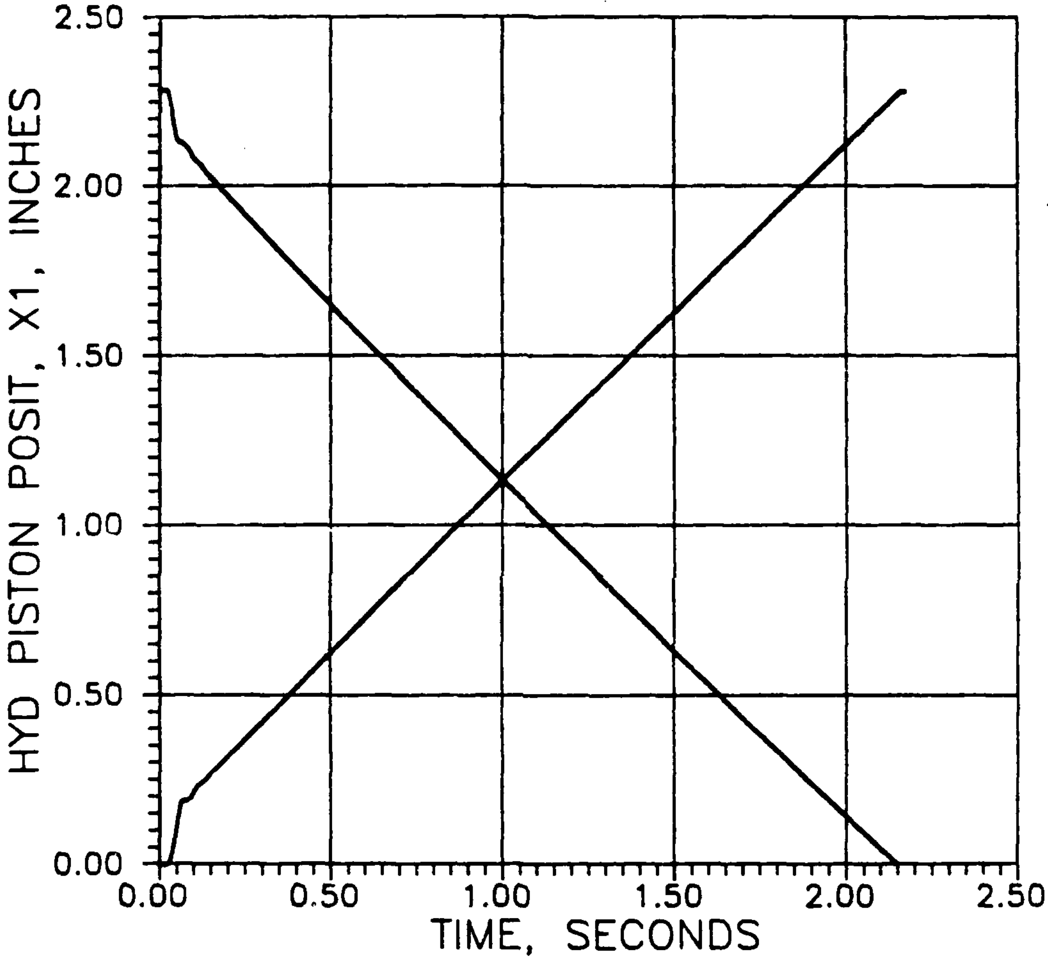


Figure F-1 No Load, Orifice 1

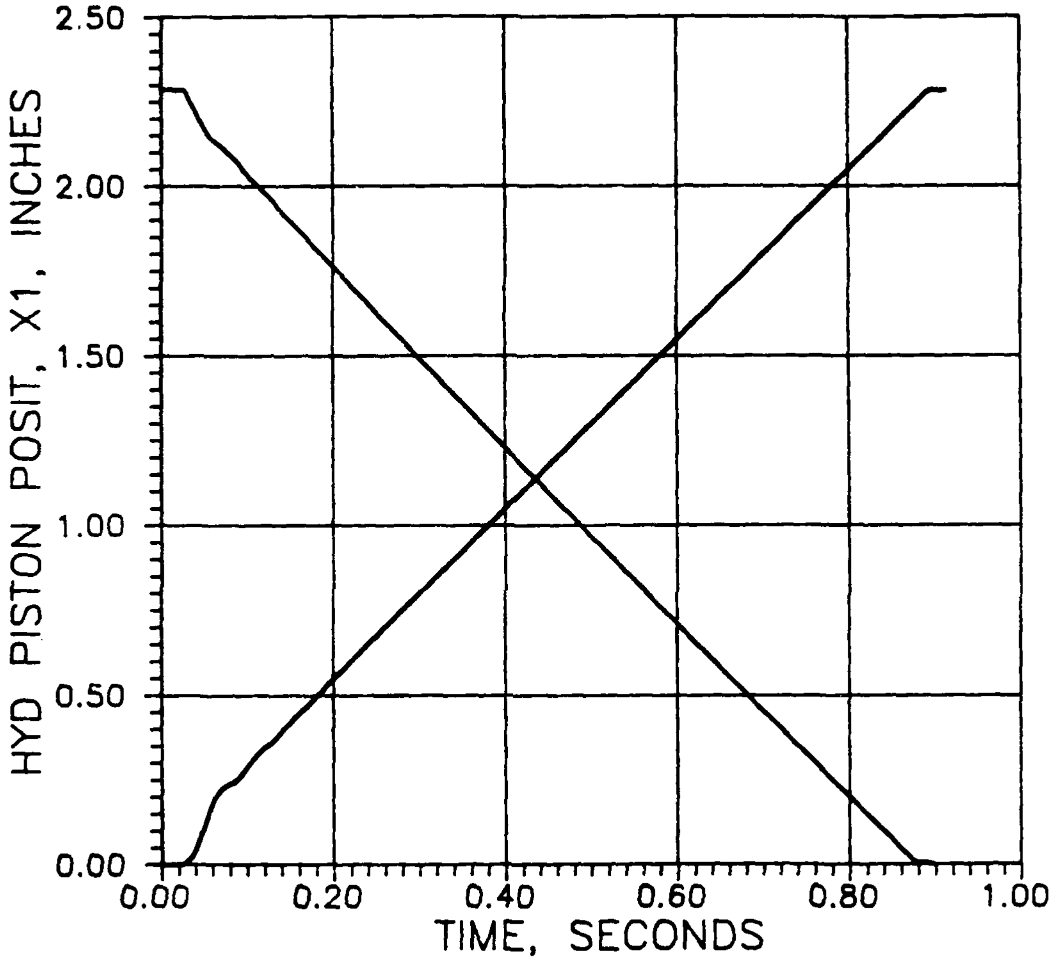


Figure F-2 No Load, Orifice 2

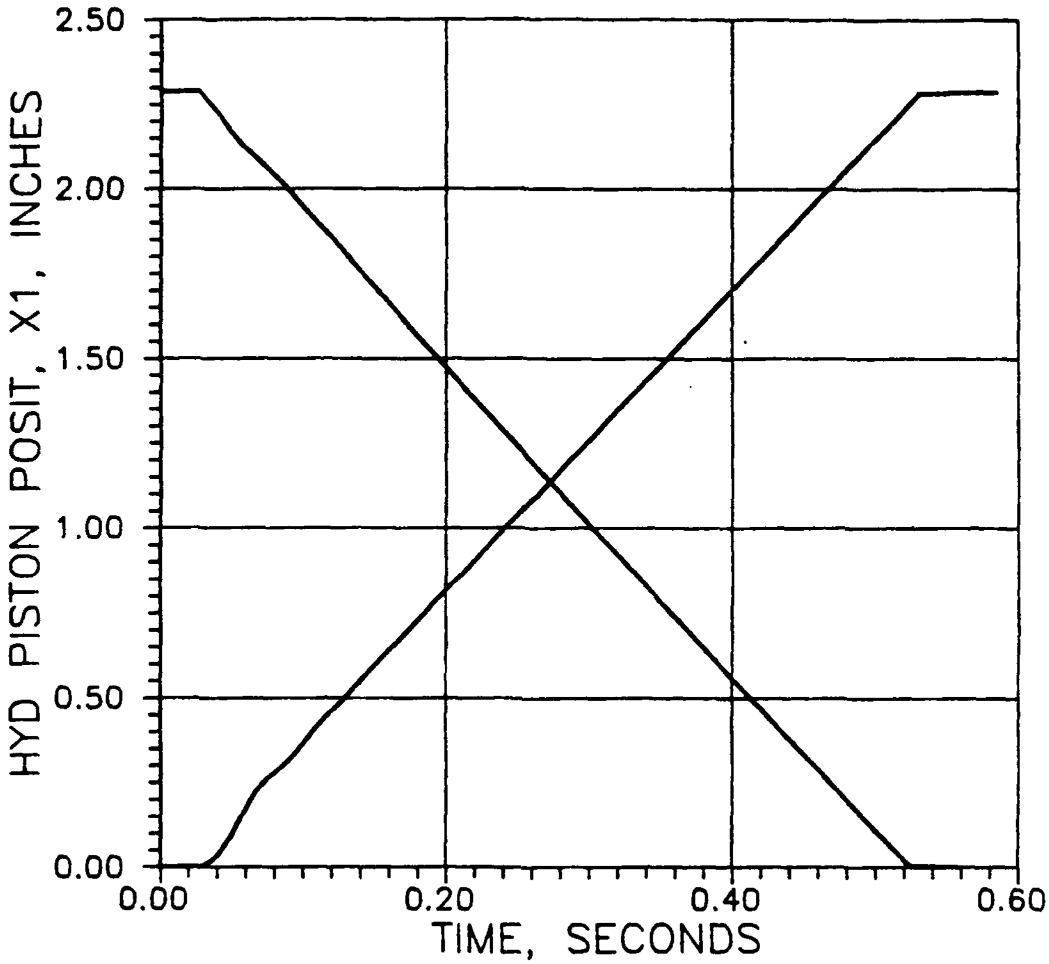


Figure F-3 No Load, Orifice 3

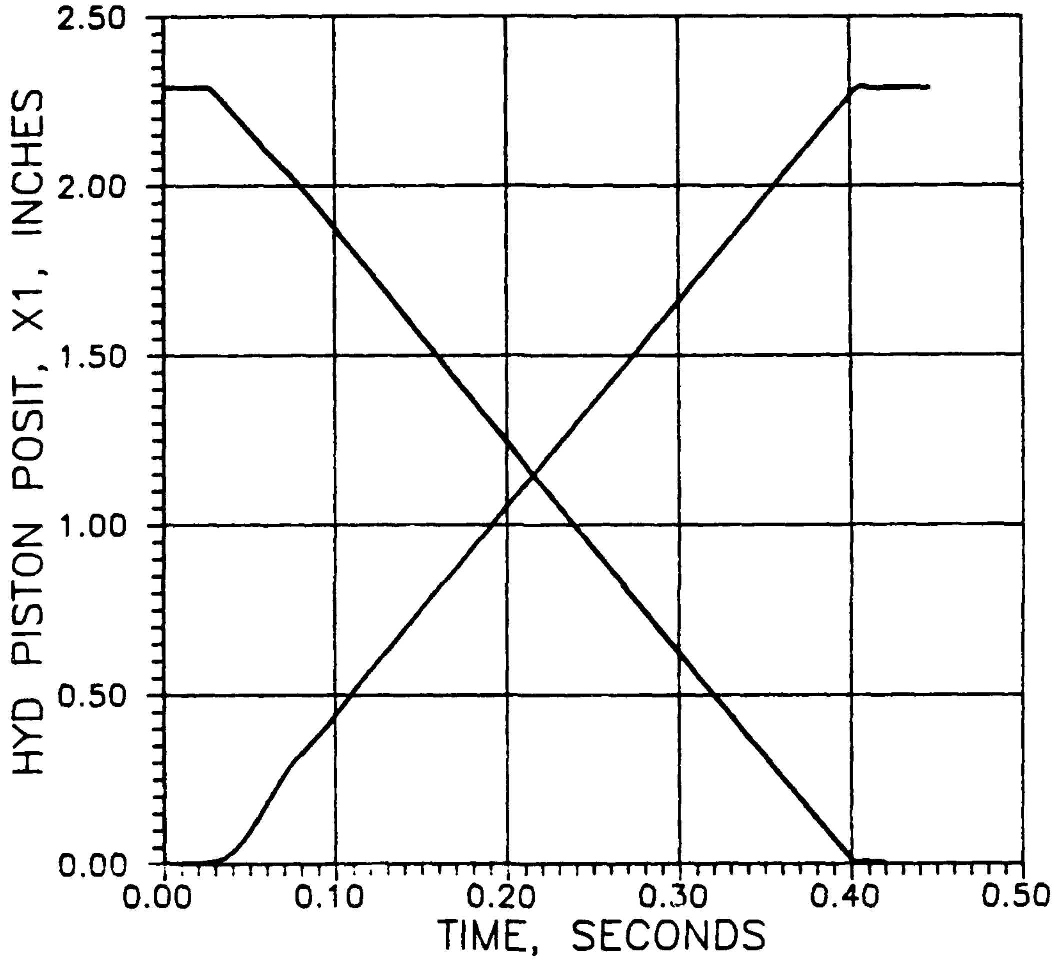


Figure F-4 No Load, Orifice 4

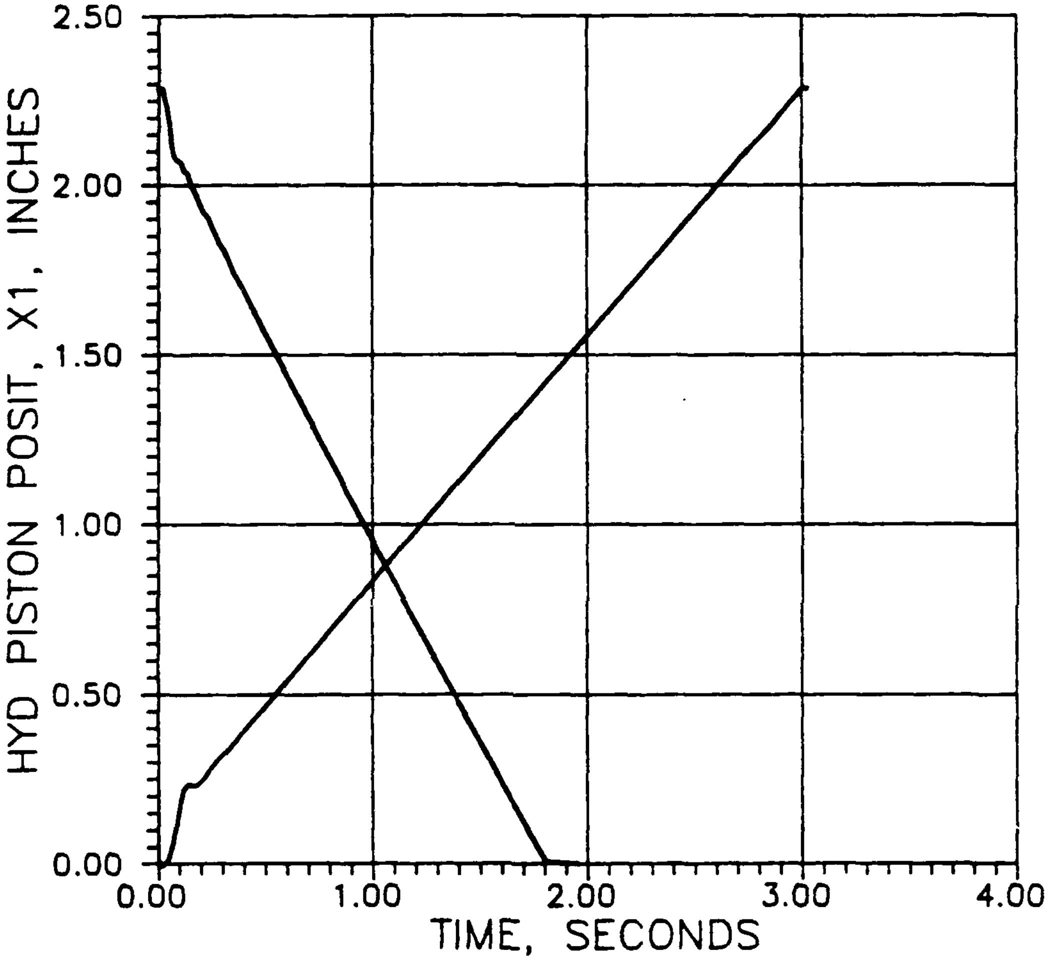


Figure F-5 2 Lb Load, Orifice 1

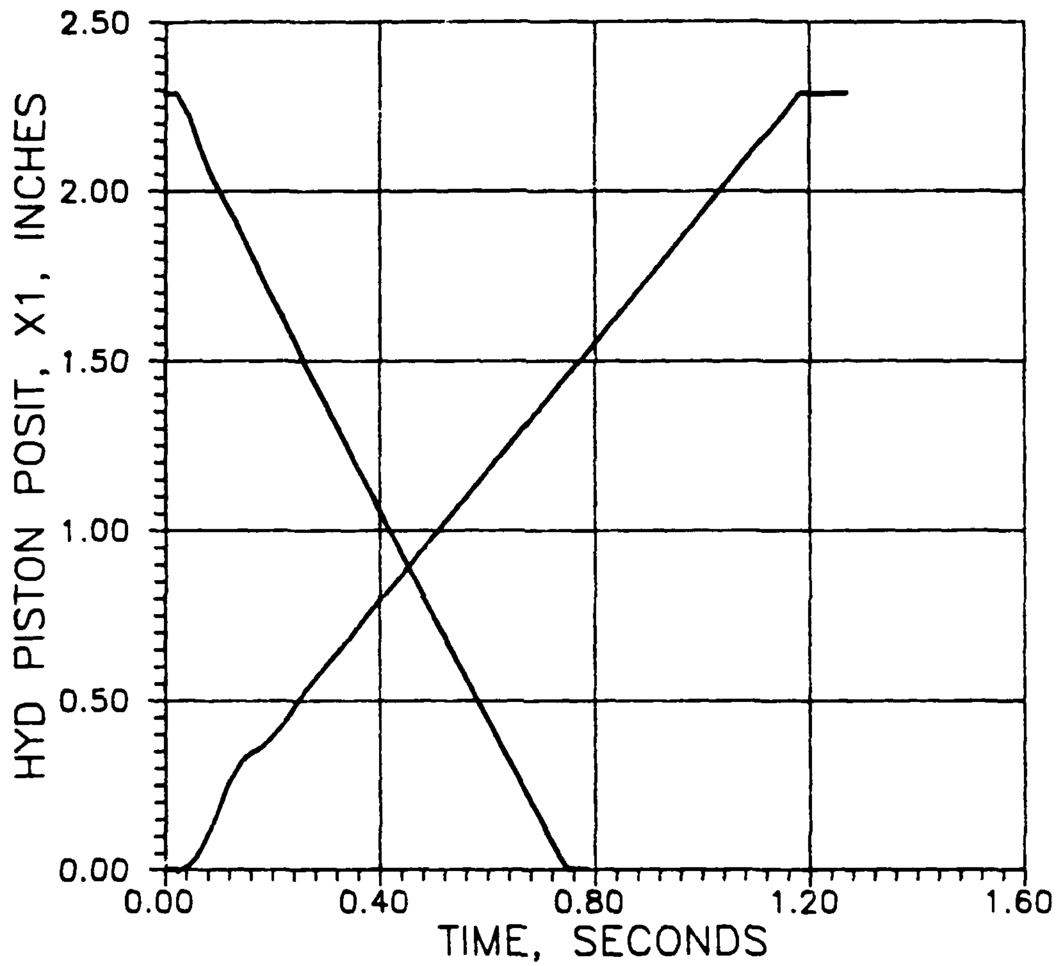


Figure F-6 2 Lb Load, Orifice 2

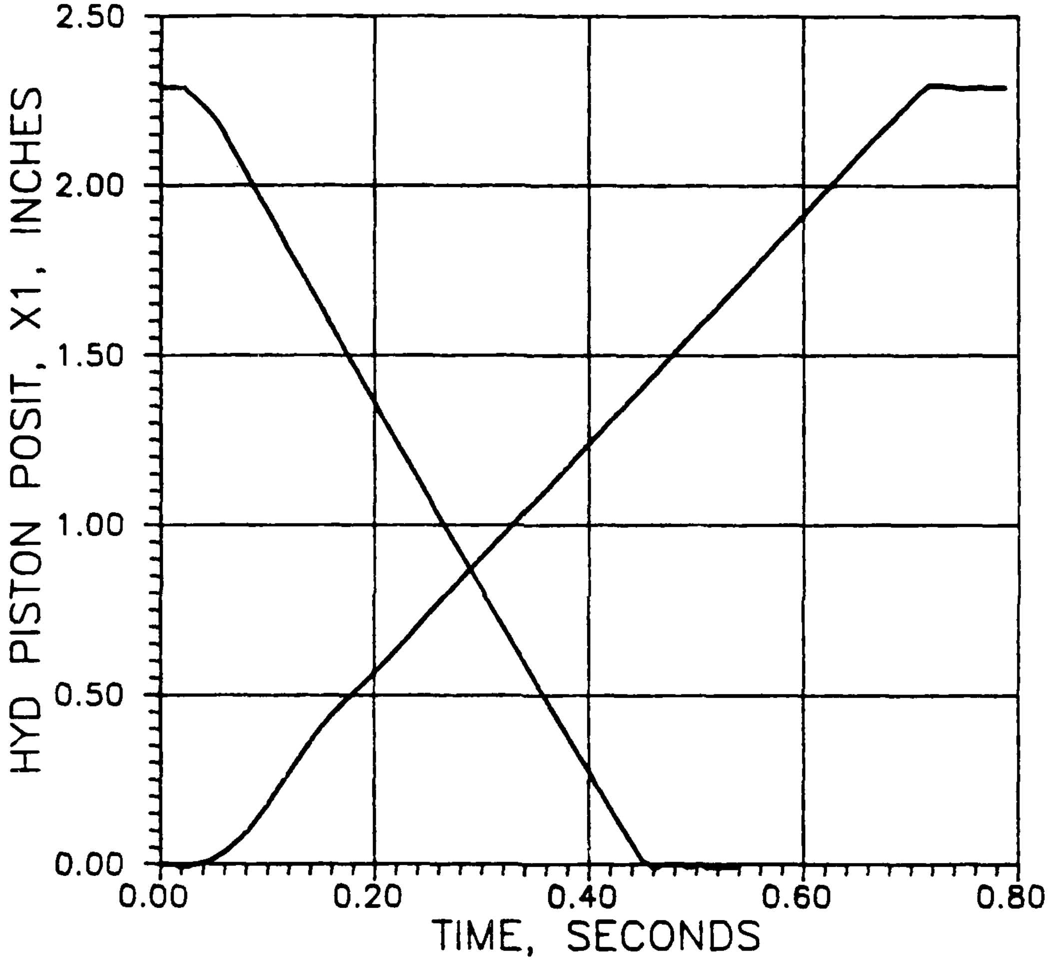


Figure F-7 2 Lb Load, Orifice 3

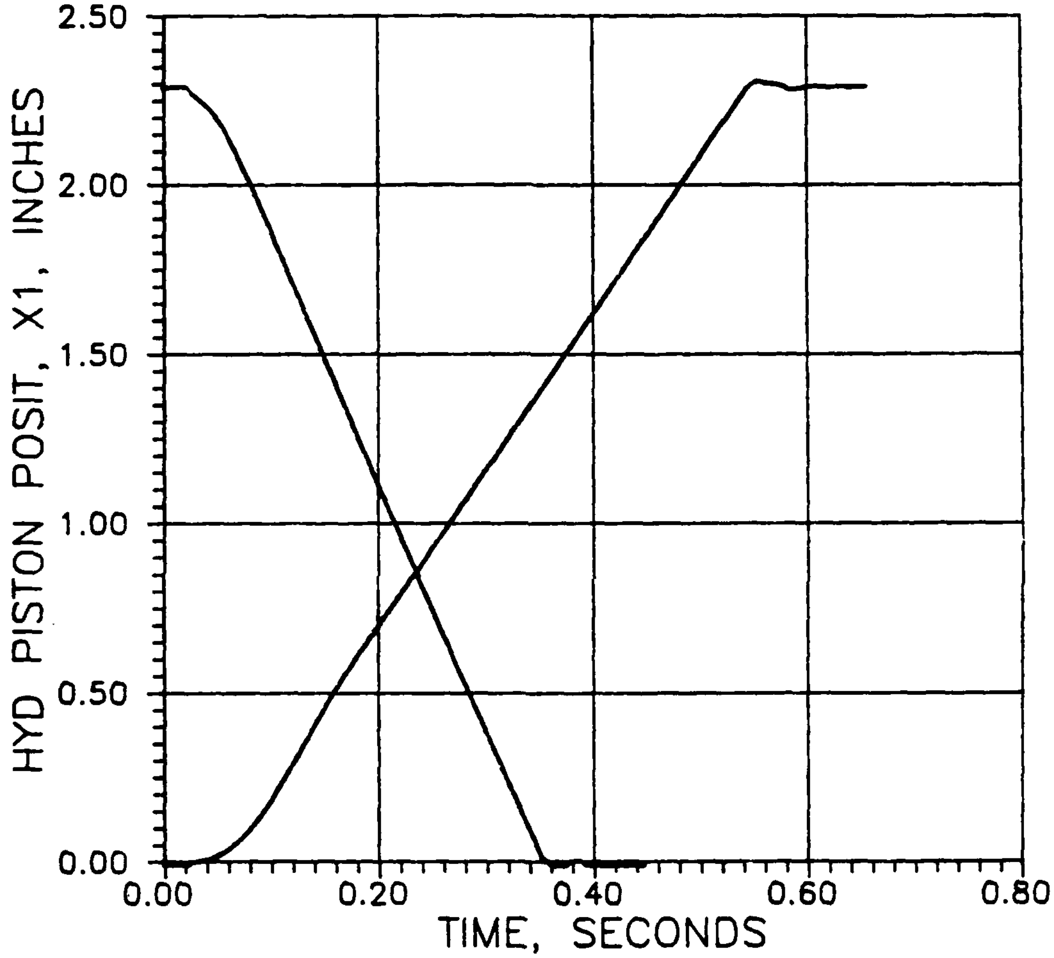


Figure F-8 2 Lob Load, Orifice 4

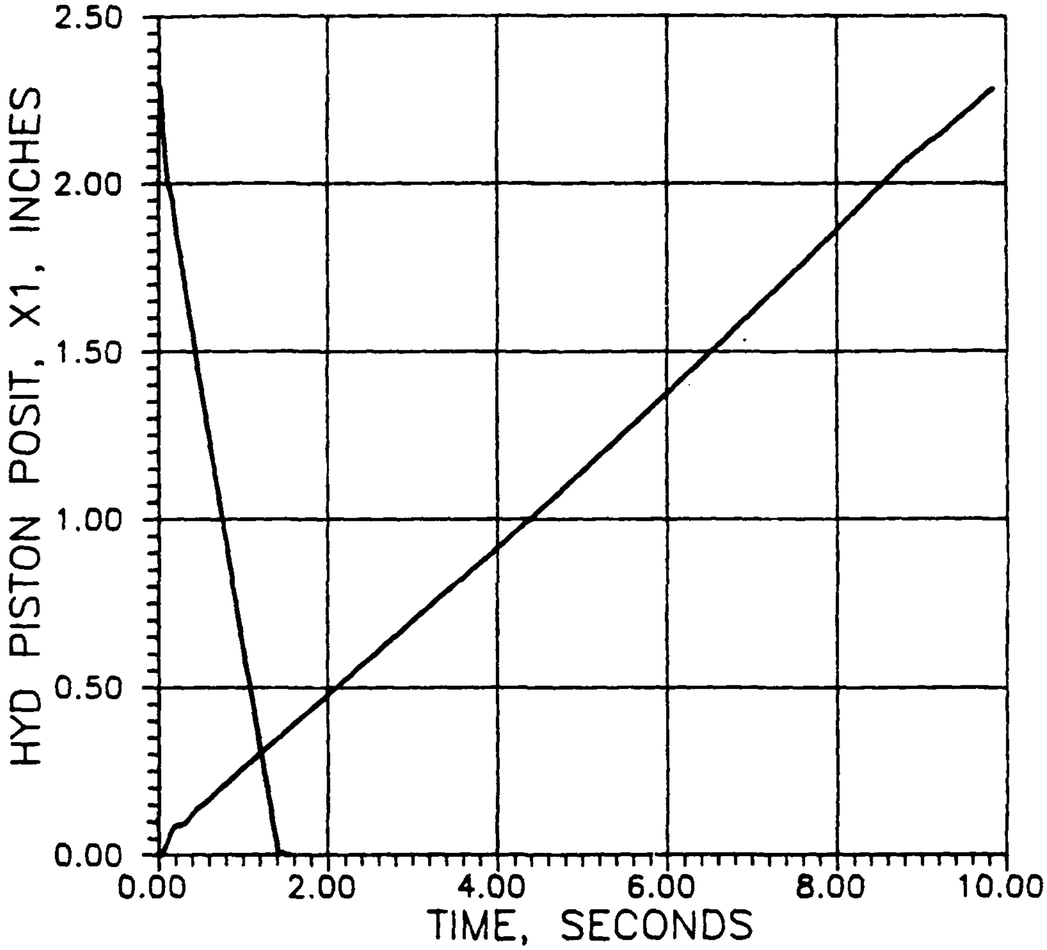


Figure F-9 5 Lb Load, Orifice 1

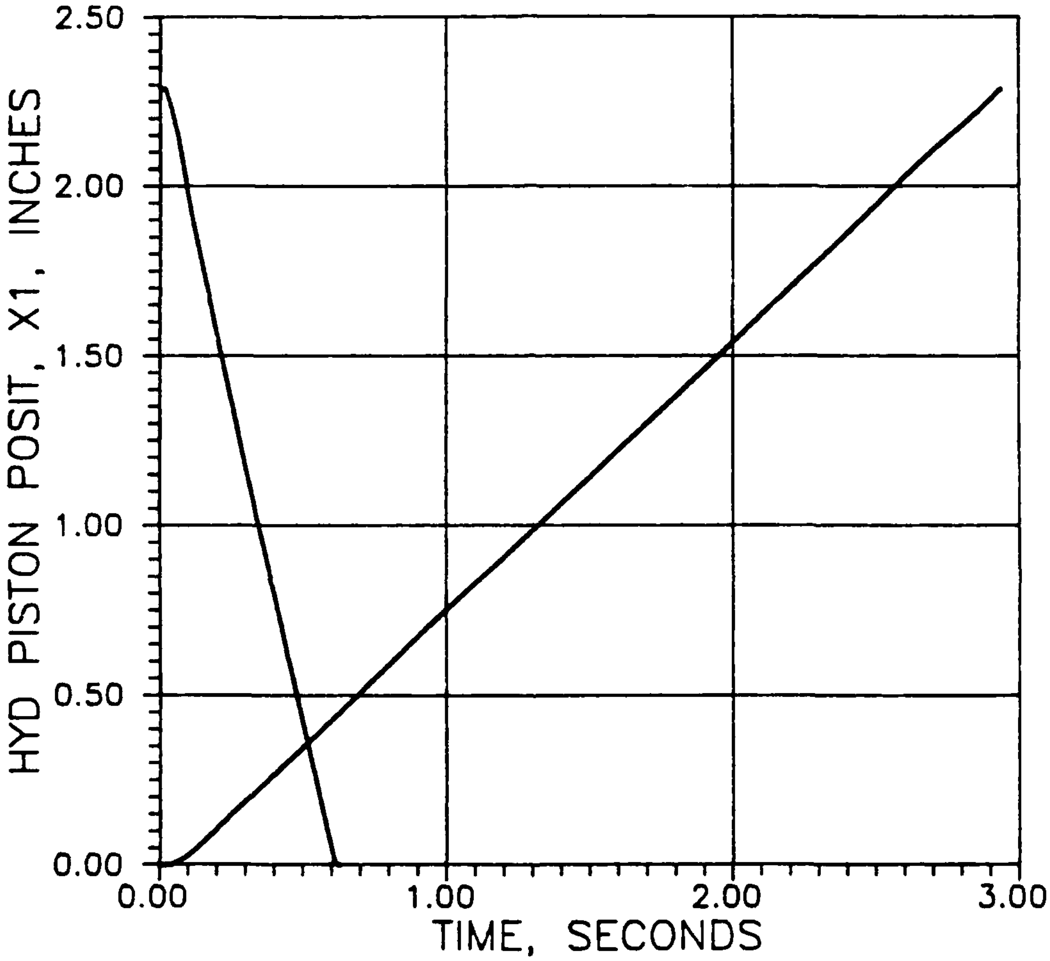


Figure F-10 5 Lb Load, Orifice 2

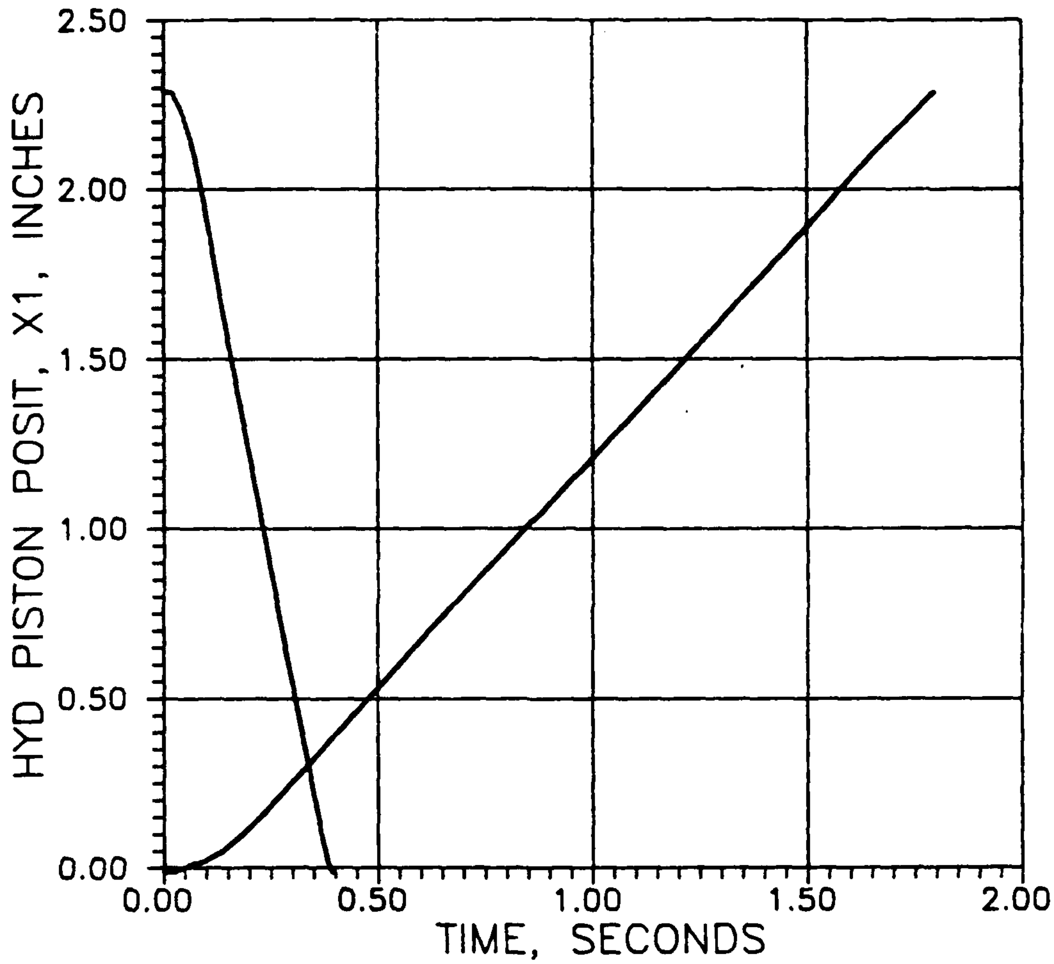


Figure F-11 5 Lb Load, Orifice 3

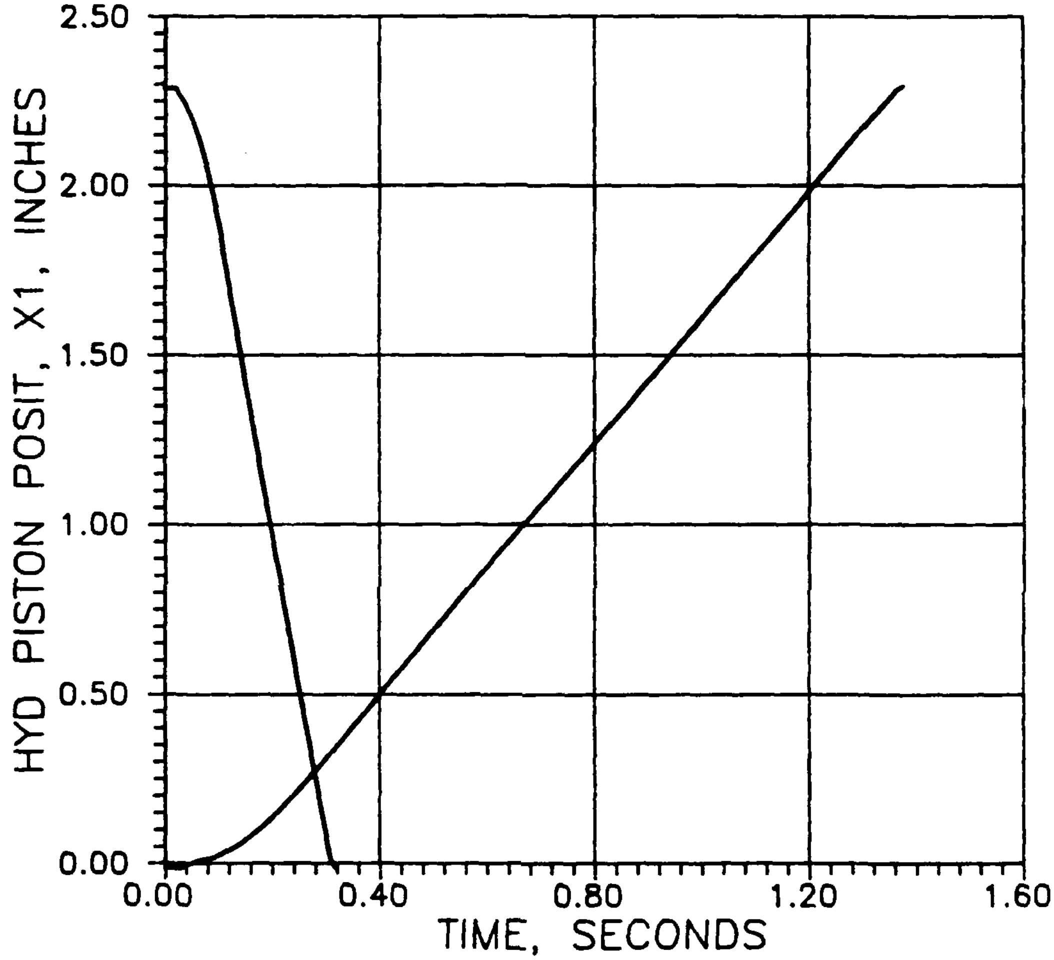


Figure F-12 5 Lb Load, Orifice 4

APPENDIX G

AUTOMATIC COMPUTER CONTROL PROGRAMS

```
'                         CONTROL.BAS
'
' THIS PROGRAM POSITIONS THE NOVEL ACTUATOR TEST BED
' MECHANISM IN RESPONSE TO A GIVEN POSITION ORDER.
'                                 JDI 11/11/88
'
'
C = &H330 : P = C + 3 : SOLENOIDS = C + 1
OUT C,0
OUT SOLENOIDS,255
OPTION BASE 1
DIM POSIDAT(1500,3)
100 OUT C,1
OUT P,1
FOR I = 1 TO 6
X(I) = INP(C)
NEXT I
XV=X(6)+X(5)*16+X(4)*256+X(3)*4096+X(2)*65536!+X(1)*1048576!
IF XV>8388607! THEN XV=XV-1.677722E+07
POSIDAT(1,2) = XV
200 PRINT "RUN AUTOMATIC CONTROLLER(1=YES,0=NO)?"
INPUT PROG%
CLS
IF PROG% = 0 THEN GOTO 800
PRINT "CURRENT POSITION IS";XV
PRINT "ENTER A POSITION ORDER FROM 0 TO 1170"
PRINT "(MUST BE AN INTEGER AND AT LEAST 200 AWAY FROM
CURRENT";
PRINT " POSITION)"
INPUT POSORD
E = POSORD - XV
MAGERR = ABS(E)
IF POSORD < 0 OR POSORD > 1170 THEN GOTO 200
IF MAGERR < 200 THEN GOTO 200
IF E > 0 THEN GOTO 300
IF E < 0 THEN GOTO 400
300 AV% = 1
GOTO 500
400 AV% = 2
'
'           INITIAL MOVEMENT ORDER DECISIONS
'
500 IF MAGERR > 724 THEN HV% = 4
IF MAGERR < 724 AND MAGERR > 374 THEN HV% = 8
IF MAGERR < 374 AND MAGERR > 187 THEN HV% = 16
```

```
IF MAGERR < 187 THEN HV% = 32
ORDER% = AV% OR HV%
NORDER% = NOT(ORDER%) AND &HFF
STARTLOOP = 1
COUNT% = 2
POSIDAT(1,1) = 0.0
MTIMER
OUT SOLENOIDS,NORDER%
DO WHILE STARTLOOP = 1
    POSIDAT(COUNT%,1) = MTIMER/1E6
    OUT C,1
    OUT P,1
    MTIMER
    FOR I = 1 TO 6
        X(I) = INP(C)
    NEXT I
XV=X(6)+X(5)*16+X(4)*256+X(3)*4096+X(2)*65536!+X(1)*1048576!
    IF XV>8388607! THEN XV=XV-1.677722E+07
    POSIDAT(COUNT%,2) = XV
    E = POSORD - XV
    MAGERR = ABS(E)
    IF MAGERR > 724 THEN HV% = 4
    IF MAGERR < 724 AND MAGERR > 374 THEN HV% = 8
    IF MAGERR < 374 AND MAGERR > 187 THEN HV% = 16
    IF MAGERR < 187 AND MAGERR > 10 THEN HV% = 32
    ORDER% = AV% OR HV%
    NORDER% = NOT(ORDER%) AND &HFF
    OUT SOLENOIDS,NORDER%
    IF MAGERR <10 THEN GOTO 600
    A = INP(&H331) AND 16
    IF A = 0 THEN GOTO 700
    INCR COUNT%
LOOP
600 OUT SOLENOIDS,255
POSIDAT(1,3) = 0.0
FOR I = 2 TO COUNT%
J = I - 1
POSIDAT(I,3)= POSIDAT(I,1) + POSIDAT(J,3)
NEXT I
PRINT
PRINT "SEND DATA TO A FILE(1) OR TO THE SCREEN(2)?"
PRINT "(IF NO DATA DESIRED, ENTER ANYTHING ELSE)
INPUT DD%
IF DD% = 1 THEN GOTO 605
IF DD% = 2 THEN GOTO 610
IF DD% <> 1 OR 2 THEN GOTO 200
605 PRINT "ENTER DOS FILENAME FOR DATA OUTPUT"
INPUT DATAFILE$
OPEN DATAFILE$ FOR APPEND AS #1
PRINT #1,"   TIME";"     POSIT"
PRINT #1,
FOR I = 1 TO COUNT%
    PRINT #1, USING "#####.#####";POSIDAT(I,3),POSIDAT(I,2)
```

```
NEXT I
CLOSE #1
610 PRINT "   TIME";"      POSIT"
PRINT
FOR I = 1 TO COUNT%
    PRINT USING "#####.#####";POSIDAT(I,3),POSIDAT(I,2)
NEXT I
GOTO 200
700 OUT SOLENOIDS,255
PRINT "CONTROL PROGRAM ABORT"
GOTO 200
800 END
```

```
'                             STOW.BAS
'
' THIS PROGRAM CAN BE USED TO PLACE THE NOVEL ACTUATOR TEST
' BED MECHANISM IN THE STOW POSITION PRIOR TO MAKING
' AUTOMATICALLY CONTROLLED RUNS OR AFTER COMPLETION OF RUNS.
'                                        JDI   11/12/88
'
CLS
C = &H330 : P = C + 3 : SOLENOIDS = C + 1
OUT C,0
OUT SOLENOIDS,255
PRINT " ENTER 1 TO COMMENCE STOW ROUTINE.  PRESS"
PRINT " KILL SWITCH WHEN ARM IS FULLY LOWERED."
INPUT A%
IF A% <> 1 THEN GOTO 100
AV% = 2
HV% = 32
ORDER% = AV% OR HV%
NORDER% = NOT(ORDER%)
OUT SOLENOIDS,NORDER%
DO WHILE A% = 1
   S = INP(&H331) AND 16
   IF S = 0 THEN GOTO 100
LOOP
100 OUT SOLENOIDS,255
CLS
END
```

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